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COMPARISON OF BOILER FEED PUMPS FOR CESIUM AND POTASSIUM RANKINE CYCLE SYSTEMS

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ABSTRACT

A study was made of the electromagnetic pump and the electric motor driven (canned rotor) and free turbine driven centrifugal pumps for the boiler feed duty in Rankine cycle systems having an output of 300 kwe with cesium or potassium as the working fluid. The polyphase helical induction pump was chosen to represent the electromagnetic pumps. The weight of the basic pump and the increase in weight of the Rankine cycle system (weight penalty) required to supply pump power and control requirements were estimated for each pump. A discussion is presented of some of the unique problems and design considerations associated with each pump such as start-up, control, bearing materials, and auxiliary equipment. Design precepts and preliminary design data for each of the pumps are presented.

For potassium boiler feed service the weight of the basic pump and the total weight of basic pump plus the weight penalty, respectively, for the helical induction pump, the canned rotor pump, and the free turbine driven pump are 397 and 660 lb, 160 and 347 lb, and 72 and 135 lb, respectively. The corresponding weights for the cesium boiler feed pumps are 1430 and 2292 lb, 274 and 697 lb, and 35 and 174 lb.

While weight is an important criterion in selecting a boiler feed pump, reliability appears to be even more important. Unfortunately, sufficient long-term operating experience has not been accumulated with any of the boiler feed pumps considered to prove reliability for the 20,000 hr or more required. An appraisal of reliability should include not only the actual pump but also the auxiliary equipment required to sustain and control pump operation. The helical induction pump has no moving parts and can be built so that it is relatively free of thermal stress problems, but it is dependent on a chain of power supply, switchgear, and control equipment and requires a cooling circuit. The free turbine driven pump requires little or no auxiliary equipment, but it inherently requires moving parts and bearings. The canned rotor pump is the most complex requiring both liquid metal bearings and electrical power supply and controls and a cooling circuit. On the bases of low weight and overall simplicity, the free turbine driven pump would appear to be the best choice to supply the boiler feed requirements of the reference design Rankine cycle systems.

(Continued)

ABSTRACT (continued)

Electromagnetic pumps were chosen for the reactor circuit (lithium) and the radiator circuits (NaK). The flat linear induction pump (FLIP) was selected for the reactor circuit, and the annular induction pump (AIP) was selected for the radiator circuit. The use of turbine driven pumps is not practicable for these applications.

INTRODUCTION

This report presents a comparison of the electromagnetic pump and the electric motor driven (canned rotor) and free turbine driven centrifugal pumps for use in Rankine cycle space power plants in which cesium or potassium is used as the working fluid. The work is a part of an analytical comparison of cesium and potassium as working fluids for Rankine cycle space power plants conducted by the Oak Ridge National Laboratory for the National Aeronautics and Space Administration.

Typical boiler feed pump requirements for 300 kwe Rankine cycle systems using potassium and cesium as the working fluids were selected from a companion report[1] that presents a series of thermodynamic cycle analyses and reference designs for cesium and potassium turbine-generator units. The efficiencies of the power turbine and generator from that report were used to compute the equivalent generator output associated with the vapor diverted to the turbine driven boiler feed pump. The various boiler feed pumps were compared with respect to the basic pump weight plus the weight penalties associated with the electrical power or the equivalent power consumed for pump operation, control, and cooling, and the weight of batteries needed for starting.

While the major portion of this report is devoted to boiler feed pumps, a section is also included on the preliminary design of a lithium pump for the reactor coolant system and the NaK pump for the heat rejection systems that are coupled to the Rankine cycle systems in the reference power systems.

BOILER FEED PUMP REQUIREMENTS

The design requirements for the cesium and potassium boiler feed pumps are listed in Table 1. It was assumed that the boiler feed pump provides both the boiler feed and the lubricant flow to the bearings in the turbine generator and in the canned rotor and turbine driven centrifugal pumps. The pump capacity was increased over the design condensate flow rate by an additional 25% and 12% for the centrifugal and electromagnetic pumps, respectively.

The additional flow would drive a jet pump to scavenge the condenser and boost the pump inlet pressure sufficiently to suppress cavitation. The possibility of suppressing pump cavitation solely by subcooling the condensate upstream of the pump suction was studied, but it was concluded that the heat rejection rate would be excessive. Cavitation suppression by means of a jet pump is much more effective; however, the large pressure drops and resultant high jet nozzle velocities might give rise to a nozzle erosion problem that should be checked in endurance tests.

Two basic boiler feed pumps were considered, the single stage centrifugal pump and the electromagnetic pump with no moving parts. Two drivers were considered for the centrifugal pump, a canned electric motor and a single-stage, partial-admission impulse turbine. Several configurations of electromagnetic pumps were reviewed, and the polyphase helical induction pump was chosen for comparison with the centrifugal pumps.

Much of the information on high-performance high-efficiency helical induction pumps was obtained from a series of General Electric Company reports on electromagnetic pump design and development[2-6] sponsored by NASA. Much of the information on the canned motor type pump was obtained from Westinghouse Electric Corporation.[7,8] The experience at ORNL was used as a basis for the preliminary design of the free turbine driven boiler feed pumps.[10]

For each pump considered, efforts were made to use the highest practically attainable efficiency and the most advanced materials to minimize the weight of each unit. For example, many helical induction

Table 1. Design Requirements for the Potassium and Cesium Boiler Feed Pumps in a 330 Kwe Rankine Cycle System

	Potassium	Cesium
Boiler outlet		
Temperature, °F	2150	2150
Pressure, psia	214.3	314.6
Enthalpy, Btu/lb	1230.5	320
Flow, lb/sec	2.21	8.79
Condenser outlet		
Temperature, °F	1330	1330
Pressure, psia	10.4	23.6
Enthalpy, Btu/lb	1049.6	274.2
Density, lb/ft^3	41.8	90.2
Flow, gal/min	23.7	43.7
Minimum ΔP		
Boiler-to-condenser, psi	203.9	291.0
Centrifugal feed pump duty		
Flow, gal/min	35.6	60.5
Head, lb/in.^{2}	234.5[a]	334.6[a]
Head, ft	807	534
Hydraulic power, kw	3.63	8.78
EM feed pump duty		
Flow, gal/min	30.54	52.94
Head, lb/in.^{2}	234.5	334.6
Head, ft	807	534
Hydraulic power, kw	3.12	7.7

[a]Includes 15% for piping friction and hydraulic decoupling between boiler and boiler feed pump.

Assumptions:

1 gal/min for each bearing (two bearings on turbine feed pump, and four bearings on turbine-generator).

25% of condensate flow added to centrifugal pump flow for jet pump to scavenge condenser and provide cavitation suppression.

12% of condensate flow added to EM pump flow for jet pump to scavenge condenser and provide cavitation suppression.

pumps with efficiencies of the order of 5% have operated in liquid metals at elevated temperatures. More recent developments[9] indicate that helical induction pumps having efficiencies as high as 20% and specific weights as little as one-fifth of earlier potassium units are already in fabrication and will soon be operated. The new higher efficiency and lower weight values were used in the study. Similarly, several free turbine driven pumps[10] with overall efficiencies in the order of 5% and weights in the order of 100 lb have been operated in potassium at ORNL. The housings for these units were fabricated from solid forgings and no effort was made to minimize weight. By selecting closer to optimum turbine wheel speeds and turbine nozzle velocities and by using reasonable care in impeller design, overall efficiencies of over 30% can conservatively be expected, and weights can be greatly reduced without sacrificing structural integrity or performance. A single stage turbine was operated in potassium in a bearing test rig at AiResearch,[11] and small multi-stage impulse turbines were operated in mercury at Thompson-Ramo-Wooldridge,[12] and Aerojet-General.[13] These turbines achieved efficiencies in the 50 to 60% range that was used for the turbine estimates in this study.

SELECTION OF BOILER FEED PUMPS

Electromagnetic Pumps

The helical induction pump (HIP) was selected as the best of the electromagnetic pumps for boiler feed service in both the potassium and the cesium cycles. Cross-sectional views of the preliminary designs worked out for these pumps are shown in Figs. 1 and 2, while data on their characteristics and performance are summarized in Table 2. The advantages associated with the HIP include the following:

1. The pump is static, with no moving parts, bearings, or dynamic seals.

2. Close tolerances are not required and differential thermal expansion is readily accommodated in the design.

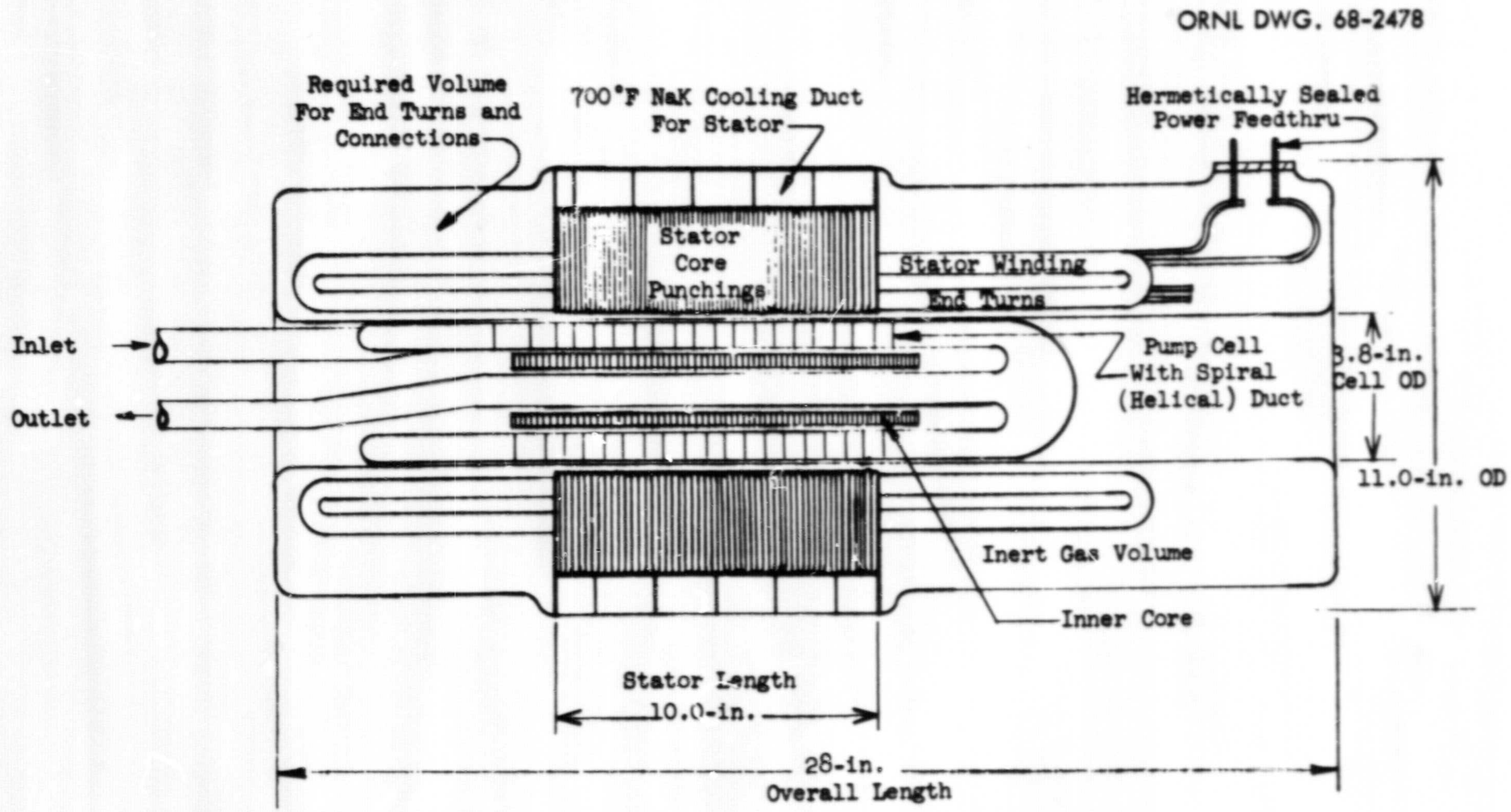

Fig. 1. Cross Section of the Preliminary Design of the Pump for Potassium Boiler Feed Application. Basic pump weight 397 lb.

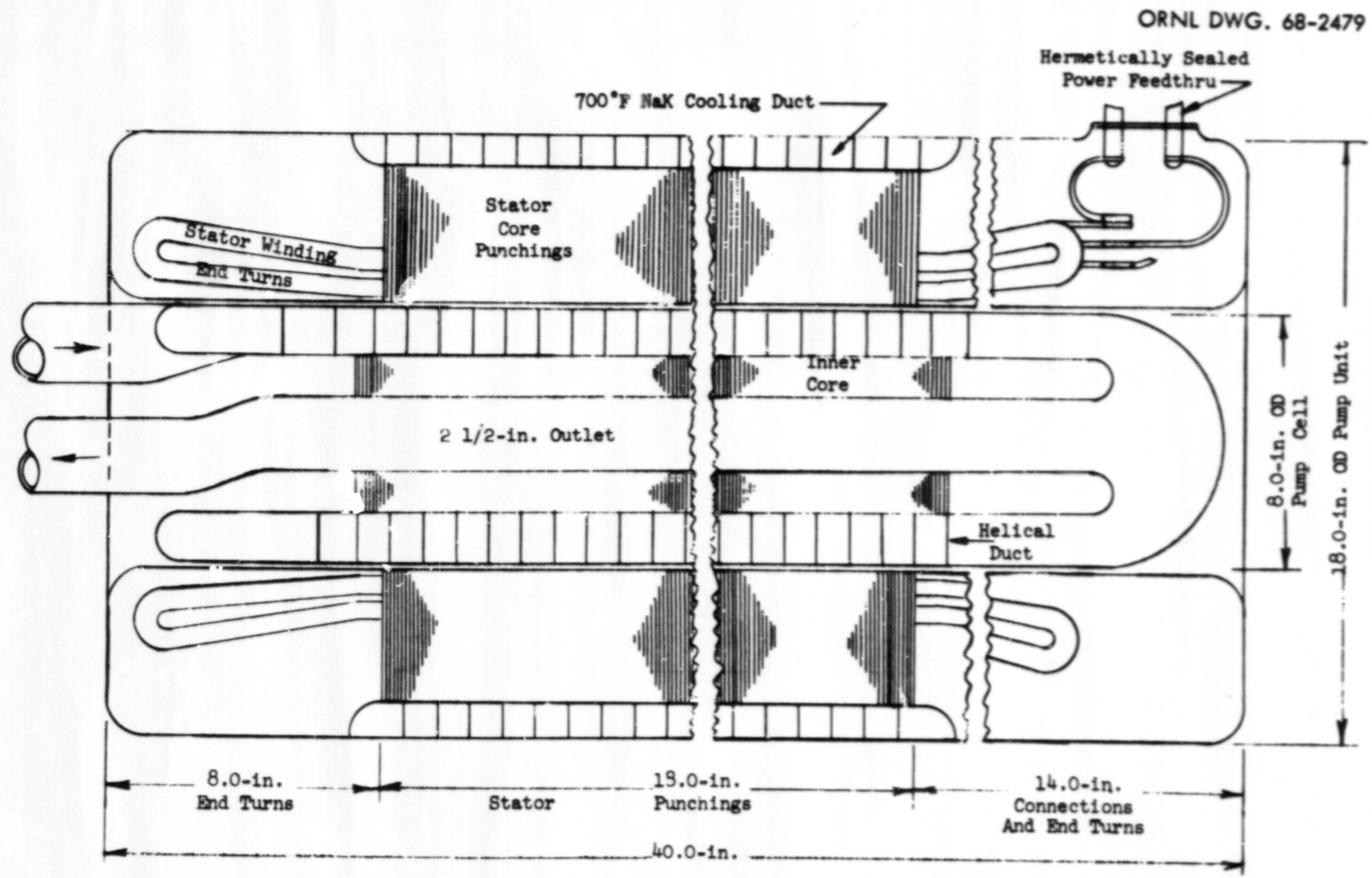

Fig. 2. Cross Section of the Preliminary Design of the Helical Induction Pump for Cesium Boiler Feed Application. Basic pump weight 1430 lb.

Table 2. Characteristics of Helical Induction Pumps for the Potassium and Cesium Boiler Feed Requirements

	Potassium	Cesium	Remarks
1. Flow, lb/sec	2.84	10.6	
2. Flow, gpm	30.54	52.94	
3. Head, psi	234.5	334.6	
4. Head, ft	807	534	
5. Temperature, °F	1330	1330	
6. Density, lb/ft^3	41.8	90.2	
7. Resistivity, microhms-in.	21.6	56.3	
8. Pump potential, volts	<400	<400	
9. NPSH, ft abs	8	6	
10. PCP $\times 10^{-3}$	155	997	
11. Pump frequency, cps	60	22	
12. Pump output, kw	3.12	7.7	
13. Pump power input, kw	16.0	51.5	
14. Pump efficiency, %	19.5	15.0	
15. Pump input, kva	34.8	129	
16. Pump P.F., %	46	40	
17. Pump reactive input, kvar	30.9	119	
18. Base pump weight, lb	397	1430	
19. Consumed power, weight penalty, lb	160	515	(10 lb/kw - Item 13)
20. Power conditioning, weight penalty, lb	16	52	(1 lb/kw - Item 13)
21. Cooling equipment, weight penalty, lb	24	77	(1.5 lb/kw - Item 13)
22. Reactive power, weight penalty, lb	23	89	(0.75 lb/kvar - Item 17)
23. Pump weight plus weight penalties, lb	620	2163	

3. The pump is of rugged construction with simplified connections and can be located in any desired orientation.

4. The system can be completely hermetically sealed.

5. Smooth flow control from zero to maximum can be obtained by simple voltage control, by frequency control, or by a combination of the two without the use of high-temperature valves.

6. Simplified startup and standby procedures are obtainable from power conditioning of battery output.

7. Plugging by oxides and particulate matter is practically impossible because flow passages are of the same order of size as the system piping.

8. The pump cell can be made of a wide range of metals and alloys.

9. The pump characteristics are well-suited to parallel and/or series operation with a single control.

10. The net positive suction head (NPSH) required is very low.

The disadvantages associated with this type of pump include the following:

1. The basic helical induction pump is heavier and larger than the free turbine pump or the canned rotor pump.

2. A separate control system and, in most cases, a separate power conditioning system are required. These can be readily combined into one system in some cases.[7,14]

3. A low power factor is associated with this type of pump.

4. The maximum efficiency is about 20%.

5. Overheating of the pump cell may occur unless corrective action is taken when the flow is stopped or materially decreased.

6. Heat generated in the stator and other components requires that the pump be cooled. In most cases a separate or auxiliary cooling system is needed. While the cooling system may be used for several electromagnetic pumps with their associated controls as well as the generator and its control system, the overall system weight, cost, and reliability are adversely affected.

Preliminary Design of Electromagnetic Boiler Feed Pumps

Electromagnetic pumps are divided into two general categories, induction pumps and conduction pumps. Further subdivisions occur in each category as shown in Fig. 3.

Conduction Pump. Studies[2 5] indicated that the dc conduction pump was superior to the ac conduction pump and therefore the latter was dropped from further consideration. Although the dc conduction pump is, in general, lighter and more efficient than the induction pump it suffers from certain disadvantages. The very high dc current required (typically thousands of amperes at less than one volt) presents problems in both power supply equipment and in the distribution of this current to and into the pump cell. Separate cooling circuits may be required for the bus bars and for the power supply equipment. These disadvantages may be overcome in some installations by proper location and design of the components.

However, the most significant disadvantages of the conduction type of pump are the attachment of the heavy electrodes to the pump cell and the resultant stresses caused by temperature gradients and physical restraints. Electrical losses also tend to cause a major heat removal problem. Because of these difficulties, the dc conduction pump was not considered further in this report. Where an overall design (including all components and effects) is to be carried out, it may be worthwhile to reconsider the dc conduction pump. A recent design of this type is discussed in Ref. 15.

Induction Pumps. Induction pumps may be classified as moving magnet or stationary magnet induction pumps. Stationary magnet induction pumps may be single-phase or polyphase. Single phase induction pumps are excited by an oscillating magnetic flux, and polyphase induction pumps are excited by a traveling wave of magnetic flux. In principle, moving magnet induction pumps differ from polyphase induction pumps only in the means of excitation. Moving magnet induction pumps are excited by the rotation of permanent magnets, or more likely, electromagnets; whereas, polyphase induction pumps are excited by a traveling wave of electromagnetic flux generated by the flow of alternating current through stationary windings.

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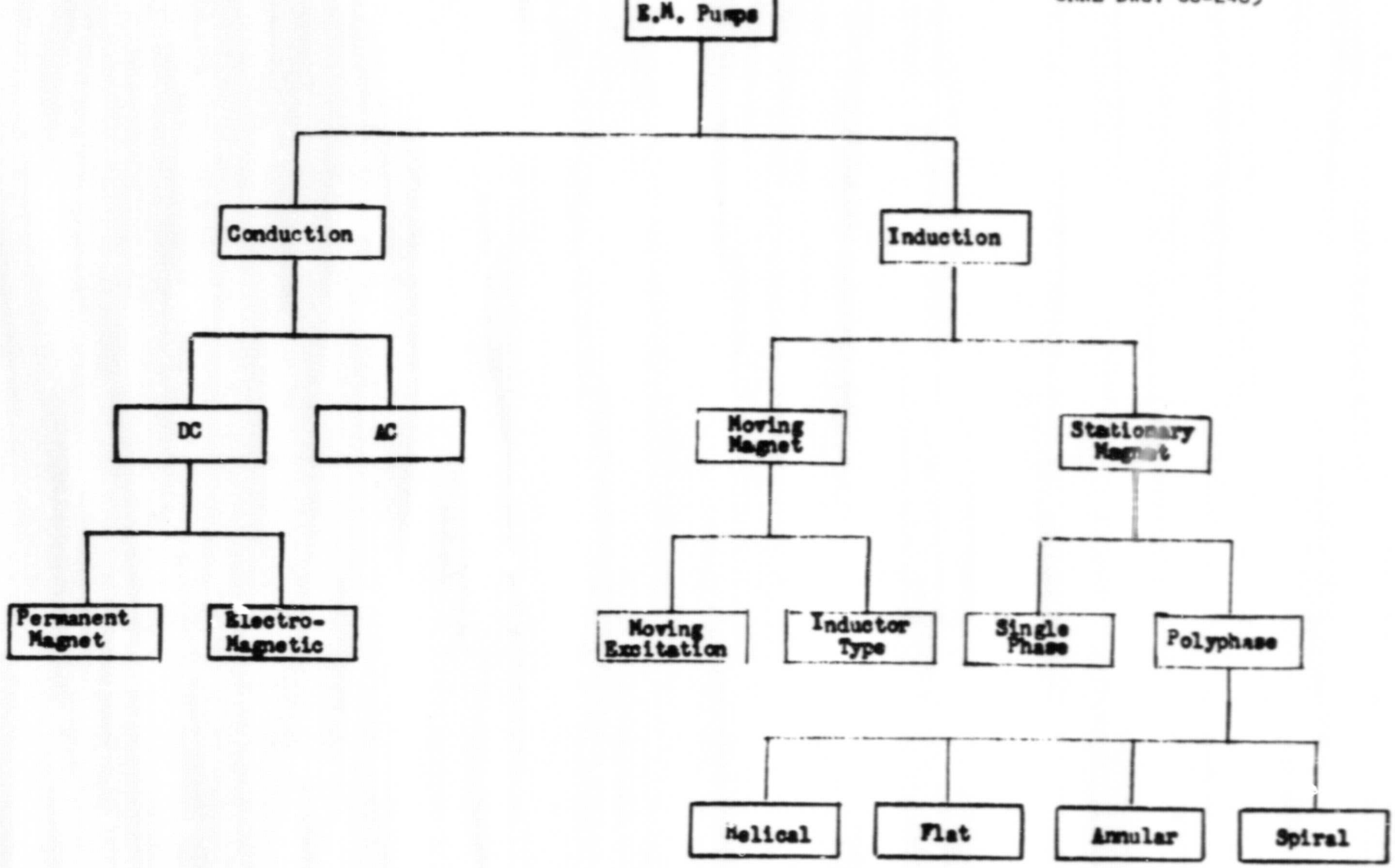

Fig. 3. Summary of Design Types of Electromagnetic Pumps.

1. Moving Magnet Induction Pumps. The electromagnet may use field windings that are either stationary or wound around the poles of the rotating structure.[16, 17] Units employing stationary field windings are similar to those used in homopolar (or inductor) induction alternators. The field windings are toroidal in shape. A traveling flux wave is produced by rotating a structure having the design number of salient poles.

In units employing rotating field windings, field current may be supplied via slip rings, or alternating power may be supplied to the rotating element by induction and to the field via rectifiers mounted on the rotating structure. The induction excitation system is now commonly used in conventional synchronous motors and generators.

This type of pump requires lower volt-ampere input and less volume of active materials than polyphase induction pumps. However, it received no further consideration because its inherent dependence on heavy rotating machinery represents complexity compared to the HIP.

2. Stationary Magnet Induction Pumps. Stationary magnet induction pumps may be single-phase or polyphase.

Single Phase Induction Pump: The principle of operation and the general arrangement of the single phase induction pump are described in detail on pages 20–22 and 65–78 of Ref. 5. However, there has been very little work reported on the single-phase pumps, and the lack of experience in design, fabrication, and operation must eliminate the single-phase pump from consideration for this application. Further, this pump seems far better suited to high or medium flow at lower or moderate head conditions.

Polyphase Induction Pumps: As shown in Fig. 3, polyphase induction pumps are subdivided into the flat linear induction pump (FLIP), annular induction pump (AIP), helical induction pump (HIP), and spiral induction pumps (SIP). Descriptions and analyses of each type of pump are provided in Refs. 2–5.

The FLIP and AIP units are both best suited to moderate and high flow, low pressure rise applications. The HIP and the SIP are best suited to low flow, high pressure applications, but in almost every comparison, the HIP is superior to the SIP. Detailed discussions of the application of these pumps are presented in Refs. 2 and 5.

The background for the above application rules can best be understood by examination of the basic equation for induction pumps and of the different pump cells used with the various pumps discussed in Refs. 1 and 4. For a given liquid metal at a given temperature, the pressure developed in a pump duct per unit length is approximated by the equation:

$$P \propto \beta^2 (V_s - V_f)$$

where P is pressure, β is flux density, V_s is the velocity (synchronous) of the magnetic wave, and V_f is the velocity of the liquid metal (consistent units must be used).

The flux density, β, is limited by the core materials to a maximum of 120,000 lines/in.^{2} and not by the type of pump. Hence, the maximum value of this parameter can be considered a constant. The fluid velocity, V_f, is limited by hydraulic losses and NPSH. The slip velocity, $V_s - V_f$, should be held within a narrow range around 0.5 V_s if the duct efficiency is to be reasonably high (Refs. 2–5).

Consequently, the development of relatively high pressures will require relatively large lengths of pump duct. This can best be accomplished in the HIP, where the long duct length can be accommodated in the helix. In order to accommodate long duct length within a reasonable pump length, it is apparent that the cross-sectional area of the duct flow passage must be reasonably small. Therefore, the flow should be in the low range.

The cross-sectional area of the flow passage for FLIP and AIP units can be made relatively large and thus these pumps are suitable for moderate to high flow applications. The development of high pressure in these pumps would require an unrealistic pump length.

Description of Selected Pump

The helical induction pump (HIP) was selected for both the potassium and the cesium cycles. These pumps are shown in Figs. 1 and 2, respectively.

Potassium Feed Pump. The design of the potassium pump follows closely the design of the flight-type helical induction pumps reported in Refs. 6, 9, and 18. This latter pump, which represents the advanced state of the art and is scheduled for completion in 1968, is designed to develop 240 psi head at 3.25 lb/sec flow of potassium at 1000°F, and has a weight of 382 lb.

A typical potassium feed pump duct material would be D43 alloy (Nb–10% W-1% Zr–0.1%C) and would have a variable pitch for the first few helix turns in order to accommodate a low NPSH. A straight center return outlet will be provided.

The stator will have a two-pole, 3 phase, 60 cps winding and will operate in a hermetically sealed argon gas enclosure. Nickel-clad silver conductors with inorganic insulation and Hiperco 27 laminations with plasma-sprayed alumina coatings will be used for the stator windings and core, respectively. The inner diameter of the stator laminations will be 4.1-in., and the outer diameter of the pump cell will be 3.8-in. The stator can and several layers of 0.002-in.-thick tantalum foil for reflective insulation will be located in the annulus between the stator and the pump cell. NaK at 700°F maximum will be used as the pump stator coolant.

The various pump characteristics and parameters are tabulated in Table 2. Weight penalties are assigned in accordance with data presented on pages 99–110 of Ref. 5. Assumptions noted in this reference have been used in the weight penalty calculations.

Cesium Feed Pump. The design of the cesium feed pump was performed using recent potassium feed pump designs[8,9] as the points of departure. The effects of differences in volume flow rise, pressure rise, and electrical conductivity were taken into account by using the curves shown in Fig. 4. These curves are based on a parameter called the pump capability parameter (PCP), which is the product of the flow capacity (gpm), head (psi), and resistivity (microhms-in.) of the pumped liquid metal at operating temperatures. It has been shown[5] that this parameter can be used as a reasonable guide to estimate the weight and size of electromagnetic pumps.

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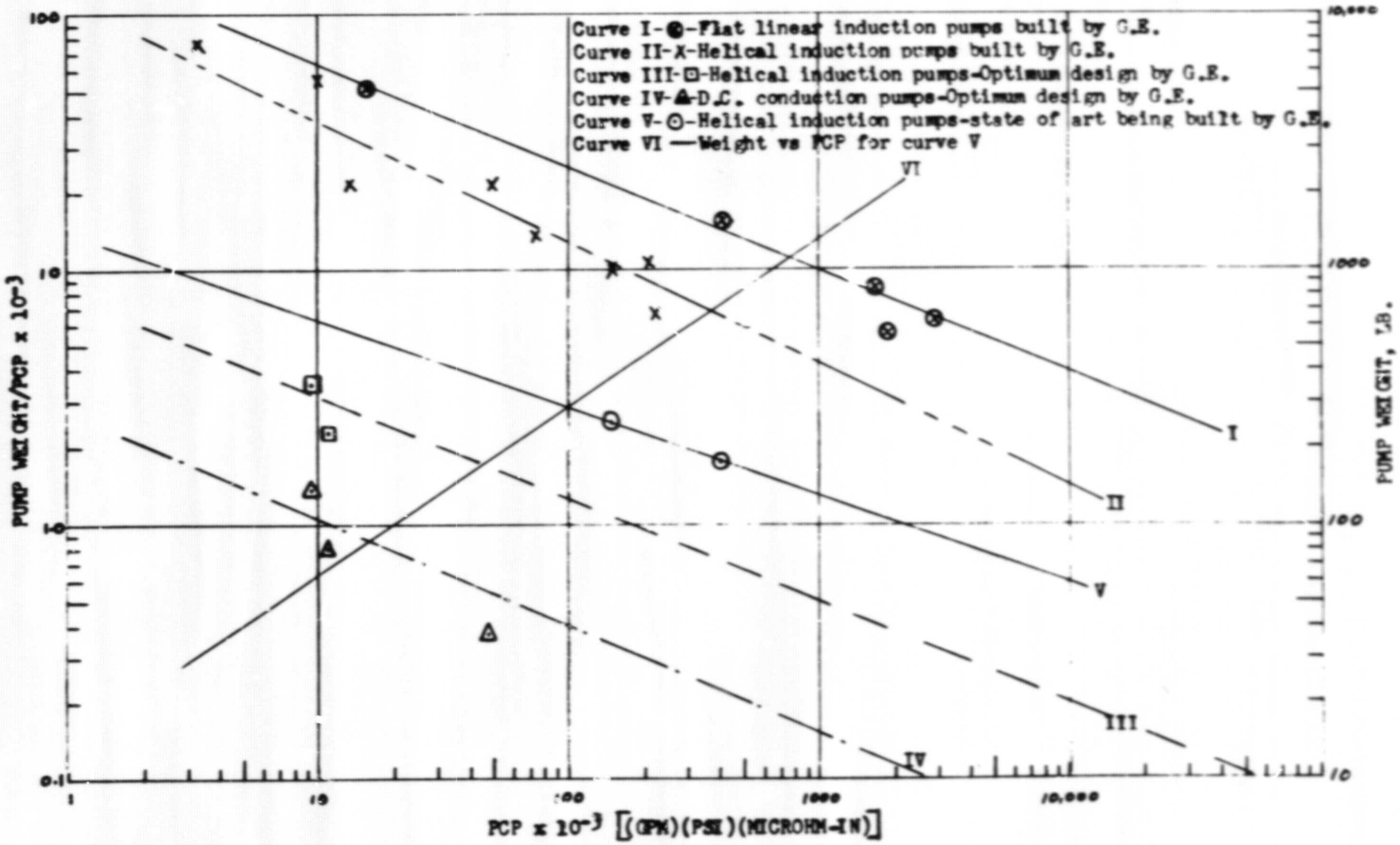

Fig. 4. Specific Weight Relationship for Electromagnetic Pumps. (Data for curve I-IV taken from Refs. 4 and 5. Data for curve V taken from Ref. 9.)

The following is quoted from page 139 of Ref. 5:

> The observed correlation of pump weight with pump capability parameter may be rationalized in the following manner. In a particular induction pumping configuration with a magnetic field of fixed peak amplitude moving at a fixed synchronous velocity, a conducting fluid passing through the pump duct at a fixed velocity less than that of the moving magnetic field experiences a pressure rise inversely proportional to its electrical resistivity. Accordingly, since the flow is the same for all fluids for the assumed conditions, the product of flow rate, pressure, and electrical resistivity is constant. Thus the pumping configuration has a capability related to the product of pressure, flow, and electrical resistivity. The weight of a pump, therefore, is a function of the parameter. This neglects significant variables such as fluid temperature, density, and viscosity. It should be employed carefully, particularly in comparing pumps for fluids with widely different characteristics or pumps designed for completely different applications or environments.

Extensive ORNL experience with Li, Na, NaK, and K is consistent with this correlation.

Curves I and II of Fig. 4 are based on pumps actually built by General Electric. Curves III and IV of Fig. 4 are based on studies reported in Refs. 2–5. On the basis of subsequent reports,[6,9,10] these two curves are beyond the present state of the art, which for helical induction pumps is represented by curve V. On the basis of personal communication with some of the authors of these reports, it is doubtful that the values of curve V can be improved by as much as 20%. Curve VI is another way of representing the data for curve V and shows that the pump weight is proportional to PCP to the 0.656 power.

The PCP for the cesium pump is 997,000 as shown in Table 2. This value of PCP has been used with curve V of Fig. 4. This operation gave a value of the ordinate (pump weight/PCP $\times 10^{-3}$) of 1.33, which in turn yields a tentative pump weight estimate of 1330 lb. However, additional pump duct changes were required by the greater hydraulic friction pressure drop with cesium compared to potassium under similar flow conditions. This greater friction pressure drop is due to the two-fold increase in density of cesium as compared to potassium and caused a 100 pound increase in the pump weight.

In view of the vital importance of the methods used to apply pump design experience to the cesium case, design of the cesium pump was approached from a different angle by using the potassium feed pump design in Fig. 1 as the starting point and by analyzing specific items in Table 2 as follows:

1. The increase in pressure rise from 234.5 to 334.6 psi for potassium and cesium, respectively, requires an increase in the length of the helical passageway by a factor of 1.43, per se.

2. The increase in electrical resistivity from 21.6 to 56.3 microhms-in. for potassium and cesium, respectively, requires an increase in the length of the helical passageway by a factor of 2.6, per se.

3. The combination of items 1 and 2 would provide an overall increase in the required length of the helical passageway by a factor of 3.72 (1.43 times 2.6) if no other parameter were changed.

4. To maintain the same fluid velocity, the increase in flow from 30.54 to 52.94 gpm requires an increase in cross-sectional area of the duct passageway by a factor of 1.73. However, this would result in an intolerable increase of approximately 50% in the hydraulic friction pressure drop per unit length of the duct passageway. Such an increase would result in an extremely steep slope of the pressure-flow characteristic curve for the cesium feed pump. This would be due to the more than twofold increase in density, whereas the equivalent diameter would be increased only by a factor of 1.31 (approximately the square root of 1.73, the change in flow). This can be seen by an analysis of the hydraulic friction pressure drop equation:

$$\Delta P = f \frac{L}{D_e} \frac{V^2}{2_g} \frac{\rho}{144} \quad \text{(in psi units)}$$

where $$D_e = \frac{2bc}{b + c} = \text{equivalent diameter}$$

with b and c being the dimensions of the rectangular duct passageway. The internal friction pressure drop for the cesium pump can be allowed to be the same percentage of the developed pressure rise as used in the design of the potassium pump. This will result in a fluid velocity of approximately 20 ft/sec for the cesium pump.

Overall analysis of slip, synchronous velocity (V_s), and frequency resulted in fixing the frequency at 22 cps, the slip at 53%, the pump cell outside diameter at 8-in., and the length of the stator punchings at 18-in. Such a design will be very close to optimum for the application. The pump dimensions and weight are shown in Fig. 2, and pertinent operating conditions are shown in Table 2.

In designing any electromagnetic pump, the problems and relations are quite complex. In preparing this section it seemed best to refer the reader to Ref. 5 rather than to attempt a repetition of that excellent presentation on which the work here is based. It is felt that the cesium pump as designed is within 10% by weight and dimensions of the optimum design using present state of the art. Computer analysis of this pump design could result in optimizing such parameters as fluid velocity, duct wall thickness, and slip.

Electric Motor-Driven (Canned Rotor) Pumps

The canned rotor pumps designed for the potassium and the cesium boiler feed applications are shown in Figs. 5 and 6, respectively. The pump characteristics are shown in Table 3. The motors are 4 pole, 400 cps, 12,000 rpm motors having metallic liners (cans) between the stator and rotor.

The main advantages of the canned rotor pump compared to the electromagnetic pump are relatively low weight, moderate efficiency, and high power factor. Flow control may be obtained by using a variable frequency cycloconverter[8] or by using a throttling valve. One disadvantage of the canned rotor pump is that, in general, it requires a separate cooling system for the stator. The temperature of the pumped liquid should be lower than the Curie point of the magnetic material used in the rotor laminations in order to provide a reasonably simple method for cooling the rotor. Additionally, being a rotating machine, the canned rotor depends on liquid metal lubricated bearings and requires a higher NPSH that the HIP.

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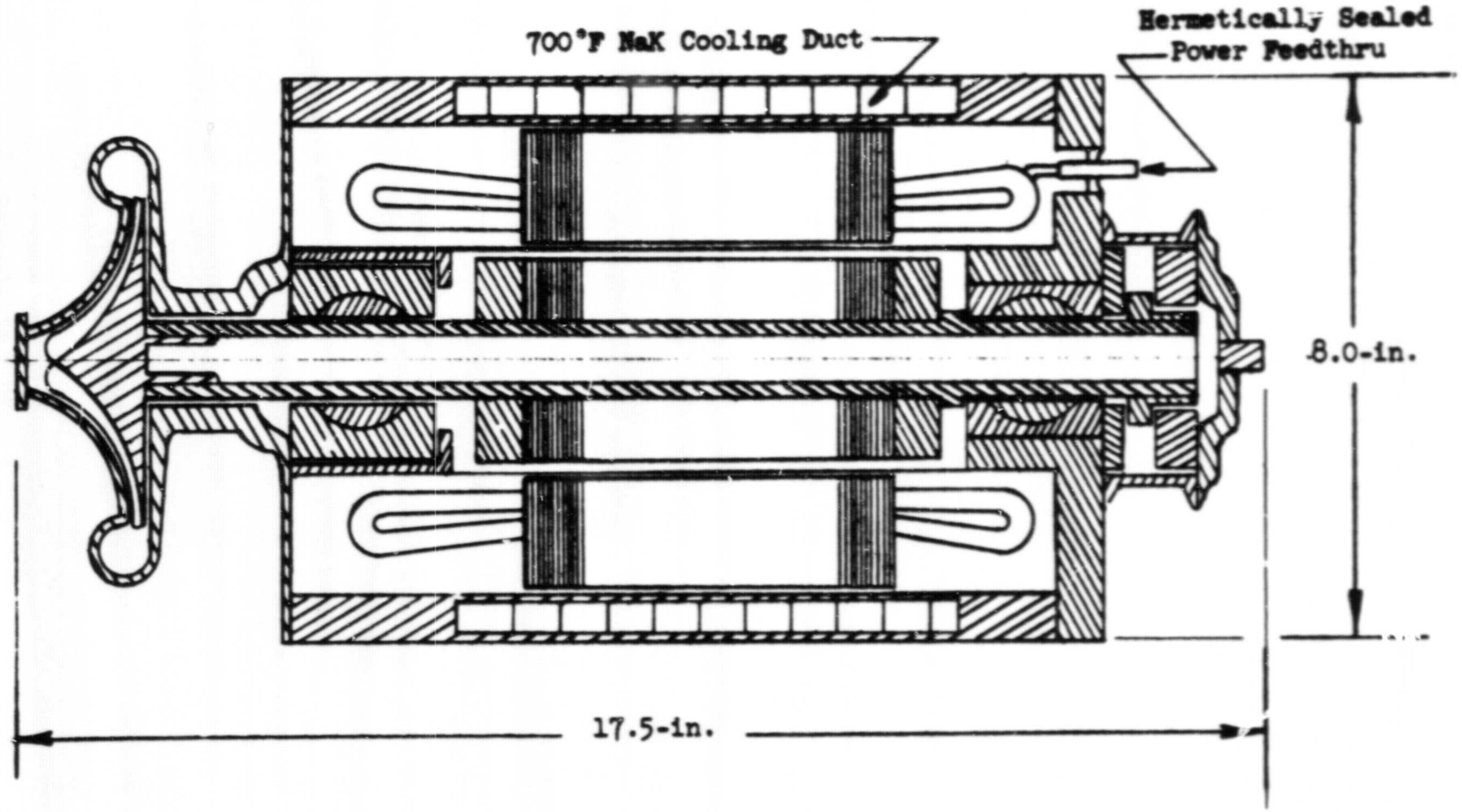

Fig. 5. Cross Section of the Preliminary Design of the Canned Rotor Pump for Potassium Boiler Feed Application. Pump weight 160 lb.

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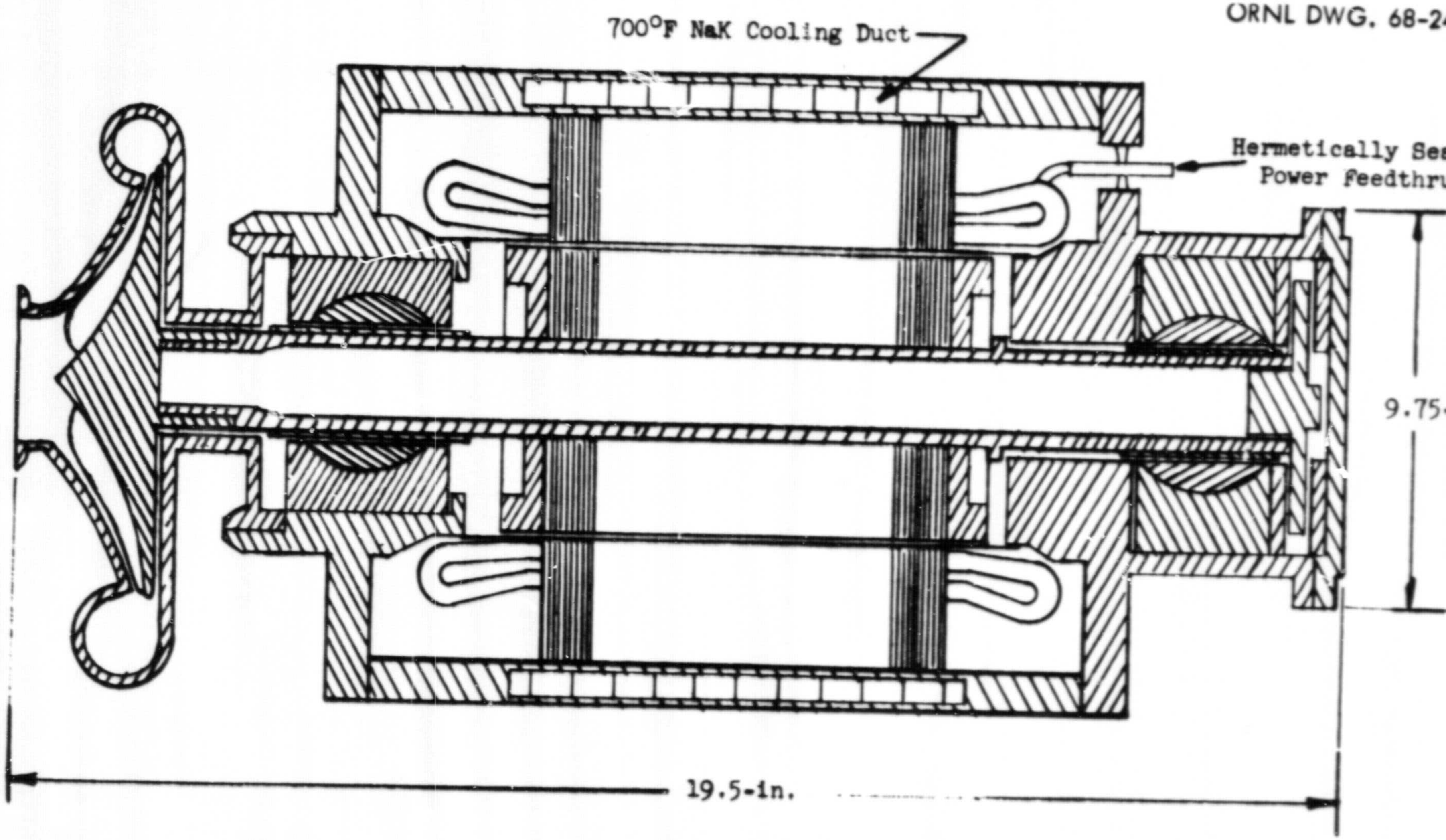

Fig. 6. Cross Section of the Preliminary Design of the Canned Rotor Pump for Cesium Boiler Feed Application. Pump weight 274 lb.

Table 3. Characteristics of Electric Motor-Driven (Canned Rotor) Pumps for the Potassium and Cesium Boiler Feed Requirements

	Potassium	Cesium	Remarks
1. Flow, lb/sec	3.32	12.1	
2. Flow, gpm	35.6	60.5	
3. Head, psi	234.5	334.6	
4. Head, ft	807	534	
5. Temperature, °F	1330	1330	
6. Density, lb/ft³	41.8	90.2	
7. Pump output, kw	3.63	8.78	
8. Pump potential, volts	<400	<400	
9. NPSH required, ft	19	27	
10. Pump power frequency, cps	400	400	
11. Pump speed, rpm	12,000	12,000	
12. Pump power input, kw	12.1	27.4	
13. Pump efficiency, %	30	32	
14. Pump input, kva	14.2	32.3	
15. Pump P.F., %	85	85	
16. Pump reactive input, kvar	7.5	17	
17. Base pump weight, lb	160	274	
18. Consumed power, weight penalty, lb	121	274	(10 lb/kw - Item 12)
19. Power conditioning, weight penalty, lb	12	27	(1 lb/kw - Item 12)
20. Cooling equipment, weight penalty, lb	18	41	(1.5 lb/kw - Item 12)
21. Reactive power weight penalty, lb	6	13	(0.75 lb/kvar-Item 16)
22. Pump weight + weight penalties, lb	317	629	

Preliminary Design of Electric Motor Driven (Canned Rotor) Boiler Feed Pumps

The canned rotor pump units considered for these applications were of conventional design using metallic bore liners (for the inside of the motor stator). The state-of-the-art for ceramic bore liners was not considered satisfactory to warrant their use for either the potassium or cesium pump. The pump rotary assembly consists of a centrifugal pump impeller, squirrel cage rotor, and bearings mounted on a single shaft. The squirrel cage rotor is covered by a 0.015-in. thick can made of D43 alloy (Nb–10% W–1% Zr–0.1% C). Whenever practical, all wetted parts of the pump other than bearings are to be fabricated from D43 alloy.

The design of the motor is discussed in general terms in Ref. 8. A silicon-iron alloy is to be used for the stator and rotor laminations, where the maximum temperature of the iron is to be less than 800°F. Laminations are to be coated with plasma sprayed alumina 1 to 2 mils thick. Heat generated in the motor is removed by liquid metal circulated through the rotor cavity and the stator cooling passages.

The motor windings are to be fabricated from nickel coated silver and insulated with double served glass.[9] This design will allow for operation at winding temperatures up to 1000°F. Slot insulation and wedges are to be high purity (99.5%) alumina shapes.

The design of the impeller and casing is similar to that for the free turbine pump operating at the same shaft speed. Detailed bearing designs were not made for either the potassium or the cesium feed pumps, but bearing losses were accounted for in the overall pump efficiency.

The limiting speed reported in a parametric study[8] was the maximum rotor speed considered. There is little reported work in this field at higher speeds. Higher speeds might be feasible, but complex stress analysis of the rotor can and rotor punchings would be required.

Potassium Feed Pump. Three speeds were investigated and resulted in the following:

rpm	Frequency	No. of Poles	Weight (lb)
3600	60	2	590
8000	400	6	210
12000	400	4	160

The efficiency of each pump unit was approximately the same, and there was no gross change in power factor. Consequently, the 4-pole, 400 cps (12,000 rpm) design was selected on a weight basis.

A 2-pole, 200 cps configuration was also investigated, but yielded a larger and less efficient pump than a 4-pole unit at the same speed.

Cesium Feed Pump. Three speeds were investigated and resulted in the following:

rpm	Frequency	No. of Poles	Weight (lb)
3600	60	2	910
8000	400	6	350
12000	400	4	274

The 4-pole, 400 cps (12,000 rpm) unit was again selected for the same reasons given for the potassium feed pump.

Free Turbine-Driven Pumps

Progress in liquid metal lubricated bearing technology in recent years makes it feasible to consider seriously the free turbine driven boiler feed pump for cesium and potassium Rankine cycle systems. Single stage relatively high-speed turbines permit low-vapor flow rates, small impellers, and compact, light-weight units. The use of the pumped fluid as the bearing lubricant provides simplicity and reliability of supply by eliminating the need for external auxiliary pumps and cooling circuitry. Several manufacturers[19,20] have recognized the advantages of the steam turbine driven pump with the bearings lubricated by the pumped condensate for high-pressure, moderate-flow, conventional marine and industrial boiler feed pump applications. Several free turbine pumps have been operated in potassium systems at ORNL for more than 8000 hr and have demonstrated basic stability and simplicity of start-up and control.

Materials specified include D43 alloy (Nb–10% W–1% Zr–0.1% C) for the housing and TZM alloy (Mo–0.5% Ti–0.08% Zr) for the turbine wheel and pump impeller. Titanium carbide with columbium binder appears to be a good choice for bearings and journals. Turbine pump shaft speeds of 12,000, 16,000, and 20,000 rpm were considered.

The pertinent aerothermodynamic design parameters for the turbines and the hydraulic design parameters for the centrifugal pump impellers were calculated, and approximate dimensions and weights of the free turbine driven boiler feed pumps were deduced. Cross-sections of the preliminary designs of three turbine driven pumps are shown in Figs. 7, 8, and 9. Comparisons between the HIP and the free turbine driven pumps were made on a power requirement basis and on weight and weight penalty bases.

Preliminary Design of Free Turbine Driven Boiler Feed Pumps

Turbine. The high pressure of the vapor available for driving the boiler feed turbine made it necessary to consider several pressure ratios and arrangements for installing the pump drive turbine in the Rankine cycle. Figure 10 shows the boiler feed turbine as a topping turbine in series with the power turbine. This arrangement utilizes high density vapor at the full flow of the power turbine with a very small pressure drop across the boiler feed turbine. This implies a very low speed and very large pump impeller. There are many undesirable features inherent in this scheme. The control of either turbine directly affects the control of the other. The boiler feed turbine wheel is at

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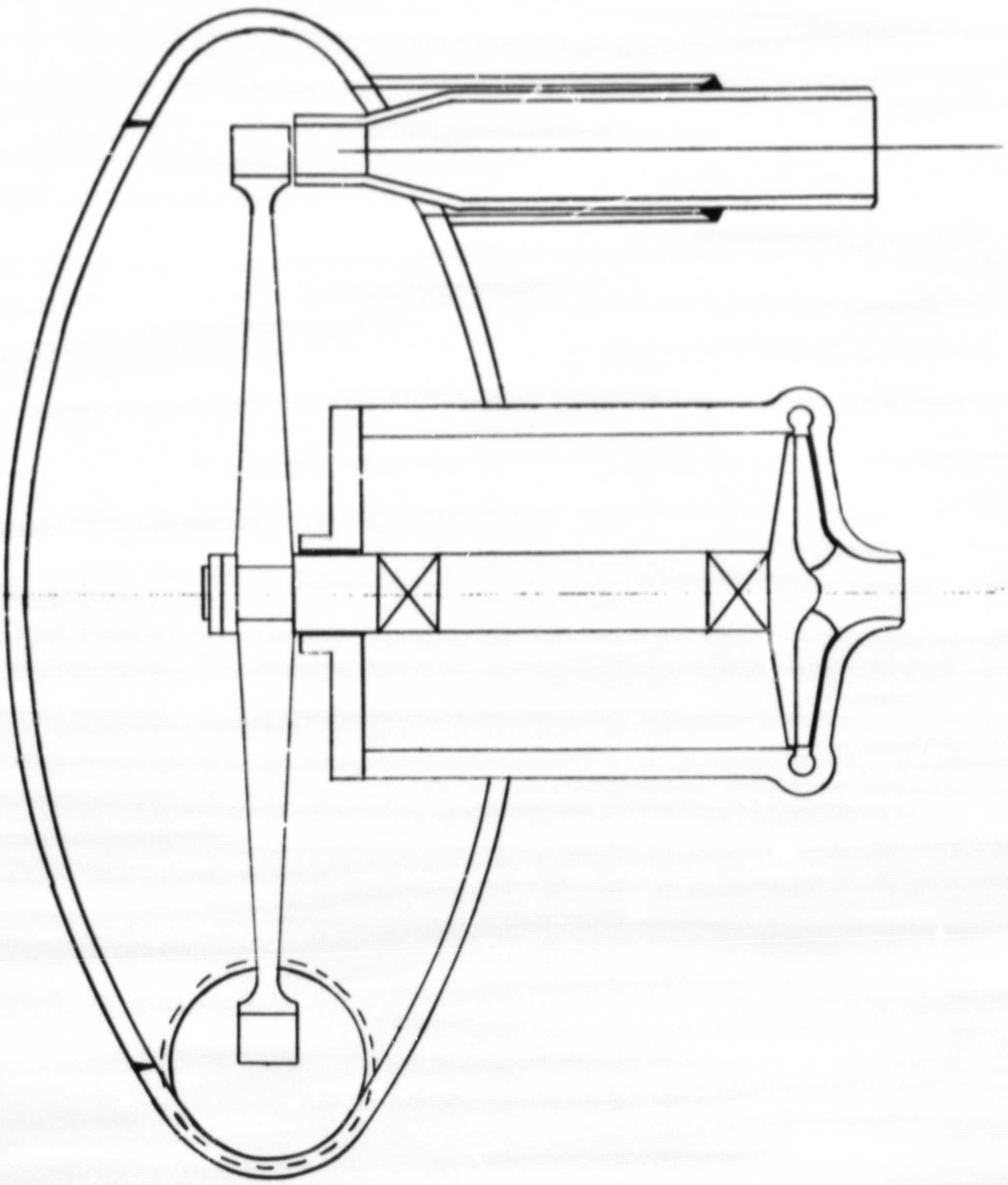

Fig. 7. Cross Section of the Preliminary Design of the Free Turbine-Driven Pump for Potassium Boiler Feed Application. Shaft speed, rpm — 12,000; Turbine wheel diam, in. — 12.0; Impeller diam, in. — 4.7; Turbine-pump weight, lb — 147.

ORNL DWG. 68-2489

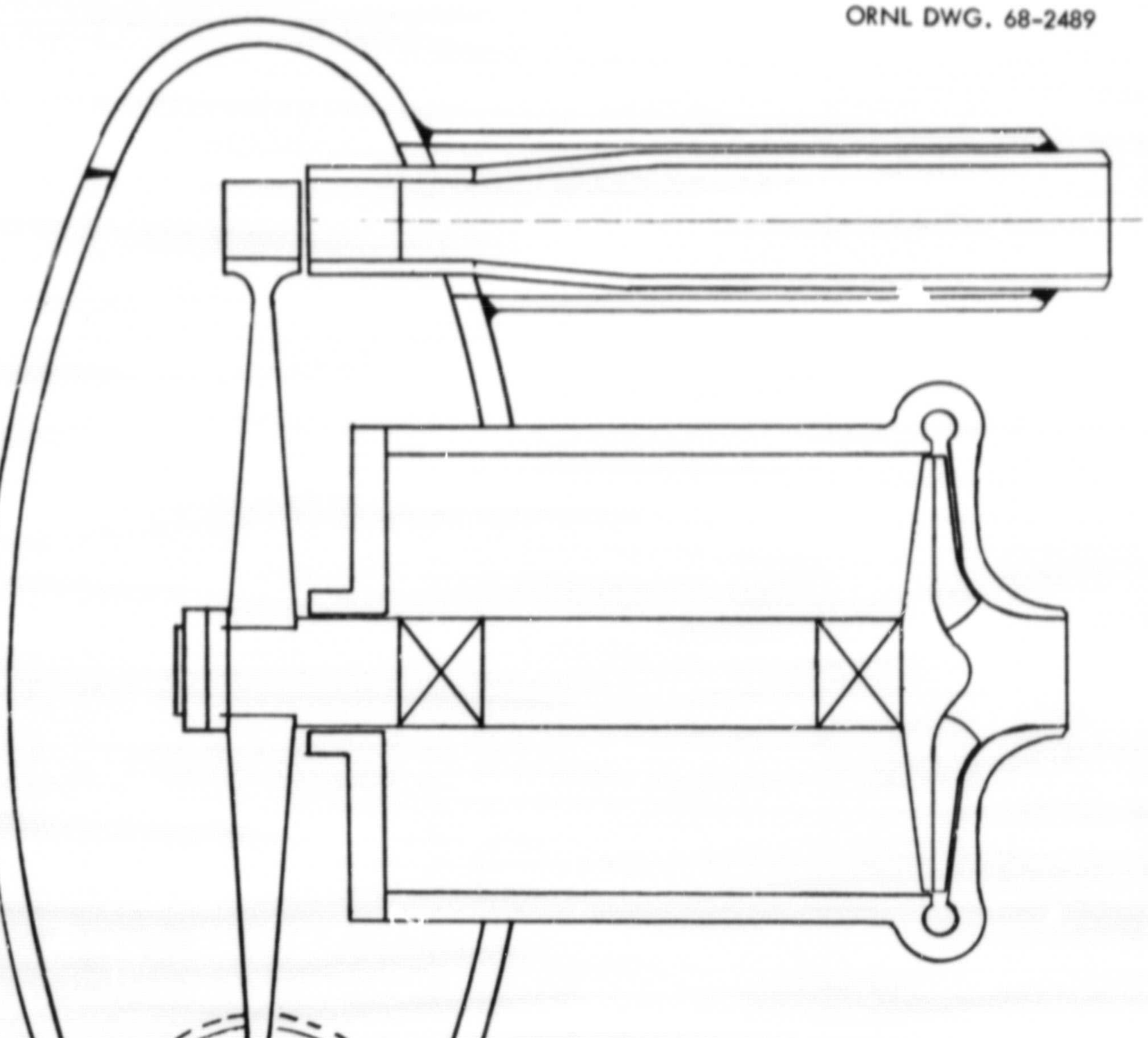

Fig. 8. Cross Section of the Preliminary Design of the Free Turbine-Driven Pump for Cesium Boiler Feed Application. Shaft speed, rpm — 12,000; Turbine wheel diam, in. — 8.6; Impeller diam, in. — 4.1; Turbine-pump weight, lb — 85.

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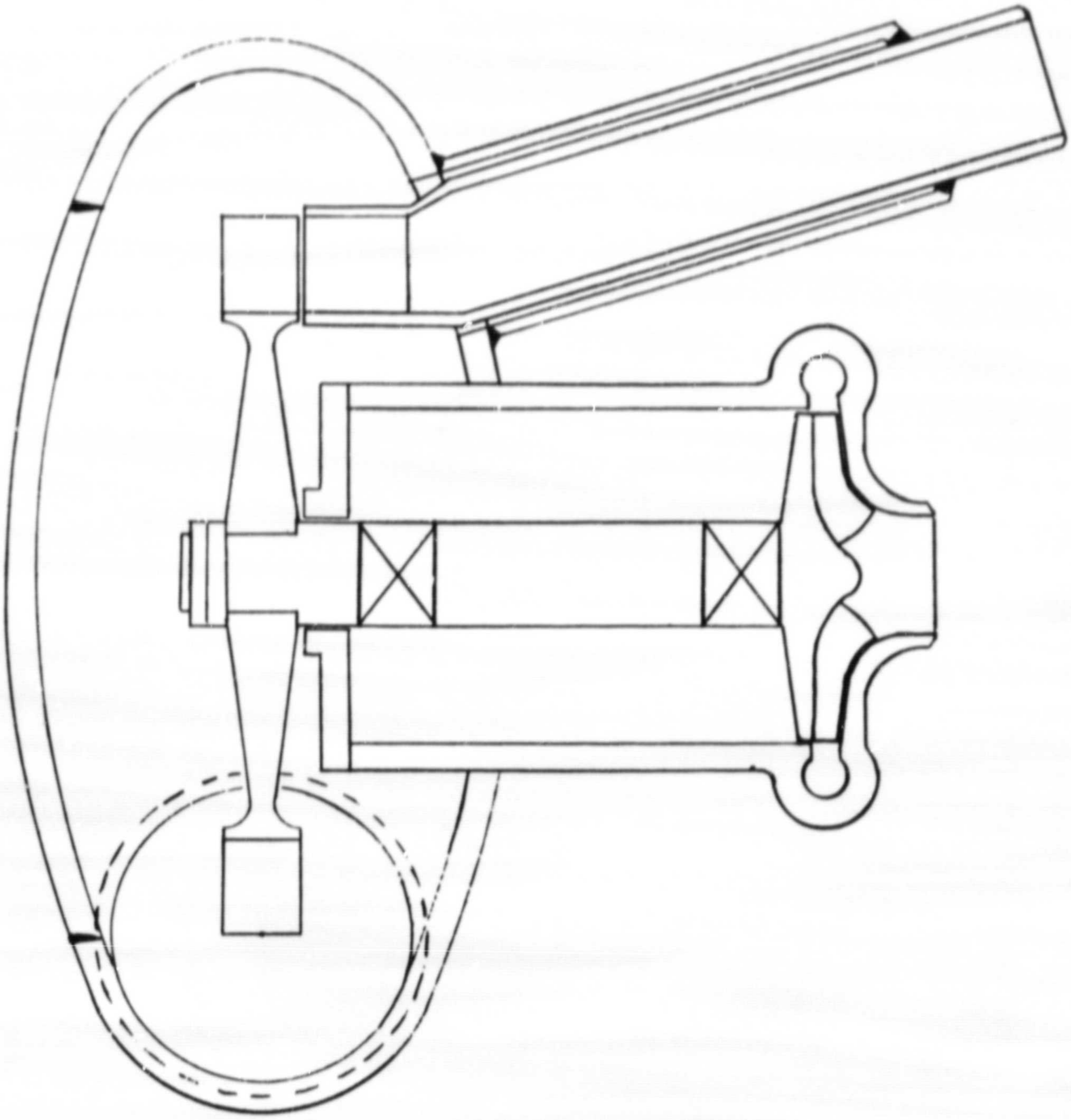

Fig. 9. Cross Section of the Preliminary Design of the Free Turbine-Driven Pump for Cesium Boiler Feed Application. Shaft speed, rpm — 20,000; Turbine wheel diam, in. — 5.5; Impeller diam, in. — 2.6; Turbine-pump weight, lb — 35.

ORNL DWG. 68-2483

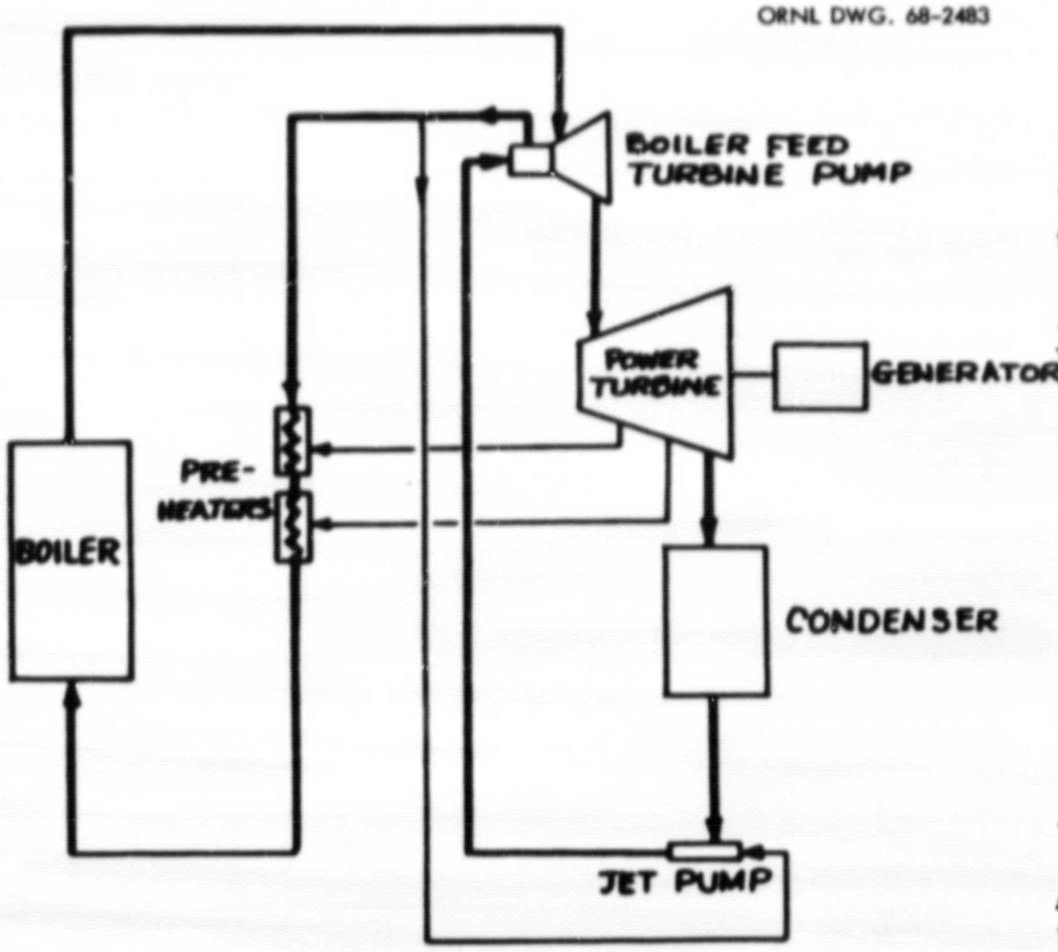

Fig. 10. Boiler Feed Turbine Pump as Topping Turbine in Ser
Power Turbine.

high-temperature so that a high-temperature gradient exists between the pump and turbine.

Figure 11 shows an arrangement in which the boiler feed turbine vapor flow is parallel to that of the power turbine. This arrangement is attractive since it provides essentially independent control of the feed turbine and the power turbine; the exhaust vapor is at very low moisture content, and the turbine wheel efficiency is relatively high. However, a single stage turbine can efficiently utilize only a portion of the pressure drop available in this arrangement. The turbine inlet pressure could be reduced by throttling at the cost of decreased availability of heat energy, an undesirable approach.

Figure 12 shows the vapor supplied to the boiler feed turbine from a stage bleed of the power turbine after full-flow moisture separation. This arrangement avoids the throttling loss shown in Fig. 11. The arrangement of Fig. 12 gives high turbine wheel efficiencies for single-stage turbines by permitting an optimization of the nozzle velocity and wheel speed. The vapor moisture content at the turbine exhaust is higher than in the parallel arrangement of Fig. 11.

Table 4 compares the pump power requirements, expressed as generator requirement in kilowatts, for the two arrangements of the turbine driven boiler feed pumps shown in Fig. 11 and 12 to that for the helical indution electromagnetic pump. Note that even though the turbine vapor flow rate is lower for the parallel arrangement, Fig. 11, the equivalent generator capacity for potassium is roughly triple and for cesium is approximately double that for the bleed turbine arrangement, Fig. 12. This results mainly from the higher availability of energy in the vapor at the turbine inlet conditions associated with the parallel arrangement. The arrangement shown in Fig. 12 was adopted and single-stage turbine wheels were sized for 12,000, 16,000, and 20,000 rpm for both potassium and cesium to supply the boiler feed duty listed in Table 1. Table 5 lists the pertinent aerothermodynamic parameters for the potassium drive turbines, and Table 6 presents comparable data for the cesium drive turbines. The turbine output requirements are based on impeller input power and bearing and windage losses. Note that a mechanical

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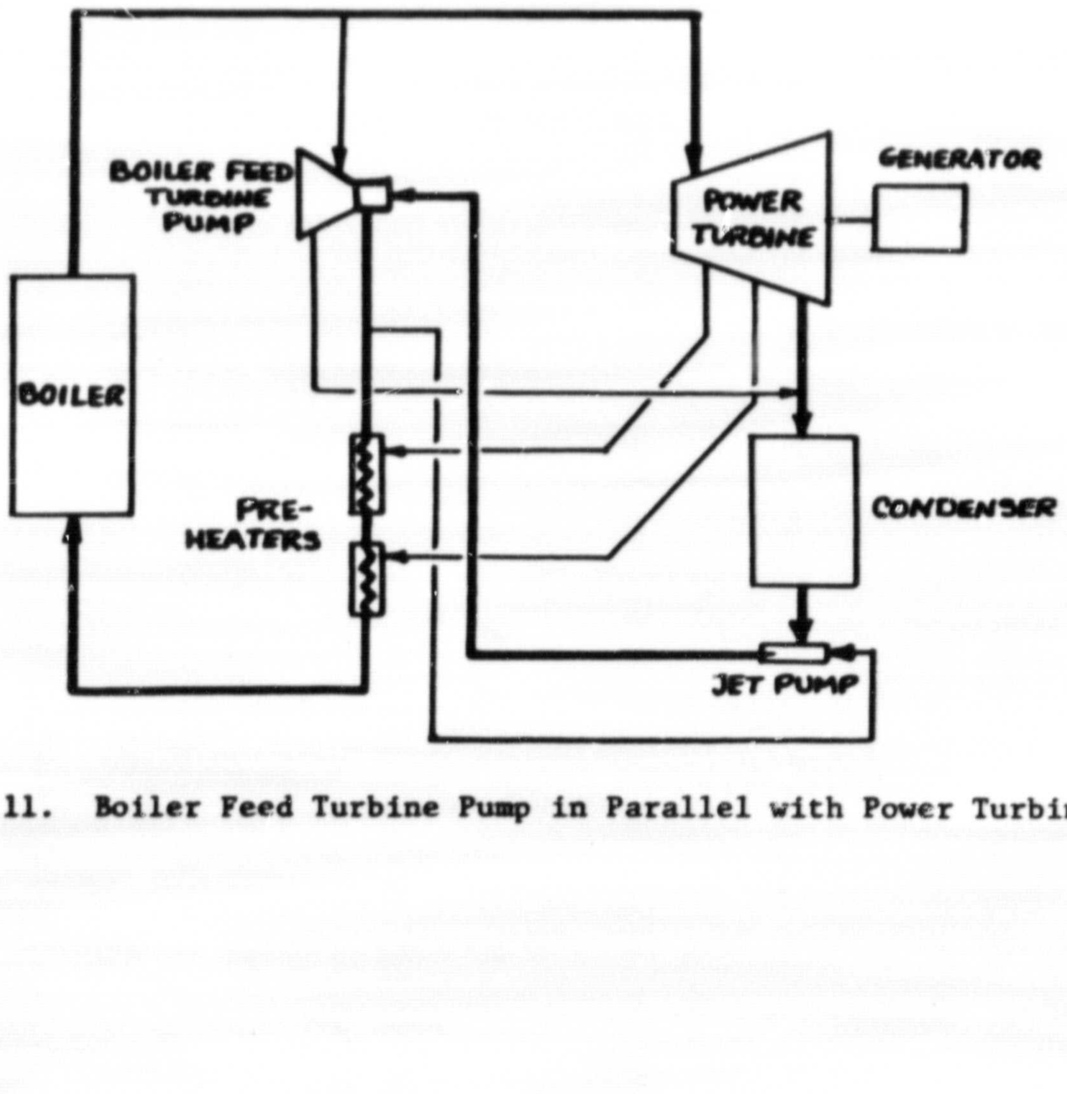

Fig. 11. Boiler Feed Turbine Pump in Parallel with Power Turbine.

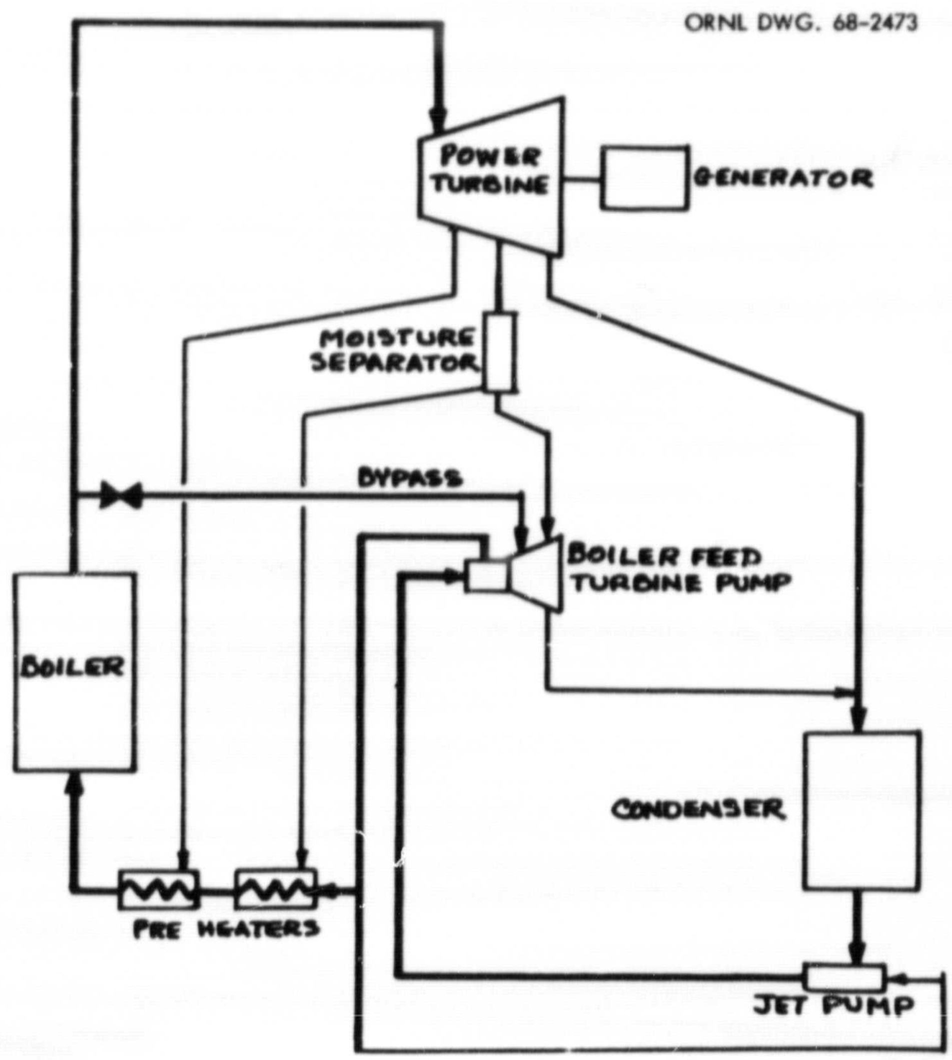

Fig. 12. Boiler Feed Turbine Pump Supplied by Inter-Stage Bleed from Power Turbine.

Table 4. Power Required by Drive Turbine for the Potassium and Cesium Boiler Feed Pumps

Power requirements expressed in equivalent generator capacity, kwe, for several shaft speeds and two turbine installation arrangements: (1) parallel vapor paths thru driven and power turbines, Figure 11, and (2) drive turbine vapor supplied with bleed from power turbine, Figure 12.

	Potassium			Cesium				
Boiler Feed Pump Drive Turbine Parallel to Power Turbine (Figure 11)								
Speed, rpm	12,000	16,000	20,000	12,000	16,000	20,000	16,000[c]	20,000[c]
Vapor flow, lb/sec	.13	.12	.12	.66	.62	.60	.66	.67
Turbine wheel efficiency, %	64	66	68	68	71	72	71	72
Power required from turbine wheel, kw	6.2	6.1	6.0	13.6	13.0	12.7	13.8	14.4
Equivalent generator requirement for the drive turbine, kw[a,b]	19.8	18.6	17.9	24.5	23.0	22.1	24.4	25.0
Equivalent generator requirement for the electromagnetic pump, kw[a,b]	16.0	16.0	16.0	51.5	51.5	51.5	51.5	51.5
Boiler Feed Pump Drive Turbine Supplied by Bleed from Power Turbine (Figure 12)								
Speed, rpm	12,000	16,000	20,000	12,000	16,000	20,000	16,000[c]	20,000[c]
Vapor flow, lb/sec	.17	.16	.15	.89	.82	.80	.87	.90
Turbine wheel efficiency, %	56.4	58.7	61.2	56.7	58.8	58.9	58.9	58.9
Power required from turbine wheel, kw	6.2	6.1	6.0	13.6	13.0	12.7	13.8	14.4
Equivalent generator requirement for the drive turbine, kw[d,e]	6.9	6.5	6.3	13.7	12.6	12.3	13.3	13.9
Equivalent generator requirement for the electromagnetic pump, kw[a,b]	16.0	16.0	16.0	51.5	51.5	51.5	51.5	51.5

[a]Generator output = 329 kw at 2.21 lb/sec K vapor flow into power turbine.

[b]Generator output = 326 kw at 8.79 lb/sec Cs vapor flow into power turbine.

[c]Flow rate of boiler feed pump increased to satisfy increased cesium jet pump requirements at two speeds.

[d]Generator output of last stage of power turbine = 69.3 kw at 1.72 lb/sec K vapor flow. From Ref. 1.

[e]Generator output of last stage only of power turbine = 104 kw at 6.77 lb/sec Cs vapor flow. From Ref. 1.

Table 5. Summary of Pertinent Design Parameters for the Drive Turbine in the Potassium Boiler Feed Pump

			Vapor Bleed from Main Turbine						Turbine Inlet				Turbine Exhaust				Turbine Parameters							
Turbine Efficiency (%)	Required[a] Turbine Output (kw)	Required Turbine Input (kw)	Bleed From Exhaust of Main Turbine Stage No.	Temperature (°F)	Pressure (psia)	h (Btu/lb)	Quality (%)	Moisture Separation Assumed Pressure Drop (psi)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Δh (Btu/lb)	Vapor Flow (lb/sec)	Exhaust Vapor Specific Volume (ft³/lb)	Exhaust Vapor Flow V_5 (ft³/sec)	H_{ad} (ft)	N (rpm)	N_S	From Fig. 1 D_S
															Without Moisture Loss									
64	6.16	9.64	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.152	43.6	6.63	46,600	12,000	9.2	5.9
67	6.06	9.06	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.143	43.6	6.23	46,600	16,000	12.6	4.5
68	6.01	8.72	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.137	43.6	6.00	46,600	20,000	15.5	3.[illegible]
															Including Moisture Loss[b]									
56.4	6.16	10.82	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.171	43.6	7.46	46,600	12,000	10.3	5.4
58.7	6.06	10.20	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.161	43.6	7.03	46,600	16,000	13.4	4.2
61.2	6.01	9.82	4	1500	24.7	1090	86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.155	43.6	6.76	46,600	20,000	16.4	3.5

[a]From Table 7.

[b]Moisture loss estimated to be 1.25% loss in power for each percent moisture at turbine exhaust.

Table 5. Summary of Pertinent Design Parameters for the Drive Turbine in the Potassium Boiler Feed Pump

		Turbine Inlet				Turbine Exhaust				Turbine Parameters															
Quality (%)	Moisture Separation Assumed Pressure Drop (psi)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Δh (Btu/lb)	Vapor Flow (lb/sec)	Exhaust Vapor Specific Volume (ft³/lb)	Exhaust Vapor Flow V_3 (ft³/sec)	H_{ad} (ft)	N (rpm)	N_S	From Fig. 13 D_S	Wheel Diameter D (in.)	Tip Speed U (ft/sec)	Nozzle Velocity C_o (ft/sec)	U/C_o	Efficiency From Fig. 13 (%)	h/D	Blade Height h (in.)	Admission (%)
								Without Moisture Loss																	
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.152	43.6	6.63	46,600	12,000	9.2	5.9	12.4	650	1729	.38	64	.055	0.68	10
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.143	43.6	6.23	46,600	16,000	12.6	4.5	9.2	640	1729	.37	67	.068	0.62	11
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.137	43.6	6.00	46,600	20,000	15.5	3.3	7.6	663	1729	.38	68	.072	0.55	14
								Including Moisture Loss[b]																	
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.171	43.6	7.46	46,600	12,000	10.3	5.4	12.0	629	1729	.36	65	.058	0.69	11
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.161	43.6	7.03	46,600	16,000	13.4	4.2	9.1	634	1729	.37	68	.072	0.66	12
86.5	0.7	24	1494	1169	96	10.4	1330	1109	90	60	.155	43.6	6.76	46,600	20,000	16.4	3.5	7.4	648	1729	.38	70	.086	0.64	15

rbine exhaust.

Table 6. Summary of Pertinent Design Parameters for the Drive Turbine in the Cesium Boiler Feed Pump

Turbine Efficiency (%)	Required[a] Turbine Output (kw)	Required Turbine Input (kw)	Vapor Bleed from Main Turbine: Bleed From Exhaust of Main Turbine Stage No.	Vapor Bleed: Temperature (°F)	Vapor Bleed: Pressure (psia)	Vapor Bleed: h (Btu/lb)	Vapor Bleed: Quality (%)	Moisture Separation Assumed Pressure Drop (psi)	Turbine Inlet: Pressure (psia)	Turbine Inlet: Temperature (°F)	Turbine Inlet: h (Btu/lb)	Turbine Inlet: Quality (%)	Turbine Exhaust: Pressure (psia)	Turbine Exhaust: Temperature (°F)	Turbine Exhaust: h (Btu/lb)	Turbine Exhaust: Quality (%)	Turbine Parameters: Δh (Btu/lb)	Vapor Flow (lb/sec)	Exhaust Vapor Specific Volume (ft^3/lb)	Exhaust Vapor Flow V_3 (ft^3/sec)	H_{ad} (ft)	N (rpm)	N_s	From Fig. 13 D_s
Without Moisture Loss																								
69	13.60	19.7	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.73	5.57	4.07	19,810	12,000	14.6	3.8
71	13.03	18.4	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.68	5.57	3.80	19,810	16,000	18.8	3.3
72	12.75	17.7	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.66	5.57	3.66	19,810	20,000	23.0	2.5
Including Moisture Loss[b]																								
56.5	13.60	24.1	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.89	5.57	4.98	19,810	12,000	16.1	3.8
58.9	13.03	22.1	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.82	5.57	4.57	19,810	16,000	20.6	3.1
58.9	12.75	21.6	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.80	5.57	4.46	19,810	20,000	25.4	2.4
Including Moisture Loss Pump Flow Increased Over Value in Table 1 to Provide Required NPSH Required for 16,000 and 20,000 rpm C_s Pump Only																								
58.9	13.80	23.4	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.87	5.57	4.84	19,810	16,000	21.2	3.0
58.9	14.40	24.4	2	1610	74.4	290	89.6	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.90	5.57	5.05	19,810	20,000	27.1	2.4

[a] From Table 7.

[b] Moisture loss estimated to be 1.25% loss in power per each percent moisture at turbine exhaust.

Table 6. Summary of Pertinent Design Parameters for the Drive Turbine in the Cesium Boiler Feed Pump

		Turbine Inlet				Turbine Exhaust				Turbine Parameters														
...ty	Moisture Separation Assumed Pressure Drop (psi)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Pressure (psia)	Temperature (°F)	h (Btu/lb)	Quality (%)	Δh (Btu/lb)	Vapor Flow (lb/sec)	Exhaust Vapor Specific Volume (ft³/lb)	Exhaust Vapor Flow V_3 (ft³/sec)	H_{ad} (ft)	N (rpm)	N_S	From Fig. 13 D_S	Wheel Diameter D (in.)	Tip Speed U (ft/sec)	Nozzle Velocity C_o (ft/sec)	U/C_o	Efficiency From Fig. 13 (%)	h/D	Blade Height h (in.)	Admission (%)
Without Moisture Loss																								
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.73	5.57	4.07	19,810	12,000	14.6	3.8	7.76	407	1125	.36	69	.08	.61	14
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.68	5.57	3.80	19,810	16,000	18.8	3.3	6.54	455	1125	.40	71	.09	.62	17
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.66	5.57	3.66	19,810	20,000	23.0	2.5	4.84	430	1125	.38	72	.12	.58	20
Including Moisture Loss[b]																									
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.89	5.57	4.98	19,810	12,000	16.1	3.8	8.61	451	1125	.40	69	.08	.69	14
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.82	5.57	4.57	19,810	16,000	20.6	3.1	6.74	469	1125	.42	72	.11	.74	18
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.80	5.57	4.46	19,810	20,000	25.4	2.4	5.14	447	1125	.40	72	.12	.74	22
Including Moisture Loss Pump Flow Increased Over Value in Table 1 to Provide Required NPSH Required for 16,000 and 20,000 rpm C_s Pump Only																									
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.87	5.57	4.84	19,810	16,000	21.2	3.0	6.66	464	1125	.40	72	.12	.80	19
[illegible]	1.0	73.4	1606	302.3	96	23.6	1330	276.8	85.5	25.5	.90	5.57	5.05	19,810	20,000	27.1	2.4	5.46	476	1125	.42	72	.14	.79	24

...xhaust.

moisture separator is used to raise the quality of the bled vapor. The first three lines on Tables 5 and 6 do not include moisture loss. Lines three thru six in these two tables include moisture loss based on 1.25% loss in turbine power for each percent moisture at the turbine exhaust.

At the higher speeds the net positive suction head (NPSH) requirement for the cesium impeller was higher than could be met with the jet pump flow assumed in Table 1. The last two horizontal lines in Table 6 show the effect on turbine vapor flow rate when the pump capacity is increased at 16,000 and 20,000 rpm to satisfy increased jet pump flow requirements. (See section entitled "Centrifugal Pump Impeller" for further discussion.)

The turbine efficiencies and wheel dimensions were determined by means of $N_s - D_s$ diagrams[21] for single-stage, partial-admission impulse turbines (see Fig. 13). The enthalpy-entropy diagrams for the potassium and cesium boiler feed turbines and also for the generator drive turbines are shown in Figs. 14 and 15, respectively.

Centrifugal Pump Impeller. Table 7 presents the hydraulic design parameters as functions of shaft speed for several centrifugal pump impellers suitable for the potassium and cesium boiler feed applications and also the characteristics of the booster jet pumps and the power requirements of the impeller, jet pump, and bearings and the windage. Two important impeller parameters are the diameter and the NPSH requirement, that is, the pump inlet total pressure above the vapor pressure of the liquid. A jet pump will be required to scavenge the condenser and to provide the NPSH requirement for the centrifugal pumps. A lower capacity jet pump will be required to scavenge the condenser and provide NPSH for the electromagnetic pumps. As noted in Table 1, it was assumed that the flow through the jet pump nozzle would be 25% and 12% of the actual condensate flow for the centrifugal and the HIP pumps, respectively. However, the 16,000 and 20,000 rpm cesium pump designs required more jet pump flow than assumed to provide the required NPSH. The last two horizontal columns in Table 6 show the increase in cesium turbine output required to provide the increased flow needed to accommodate the higher NPSH requirement.

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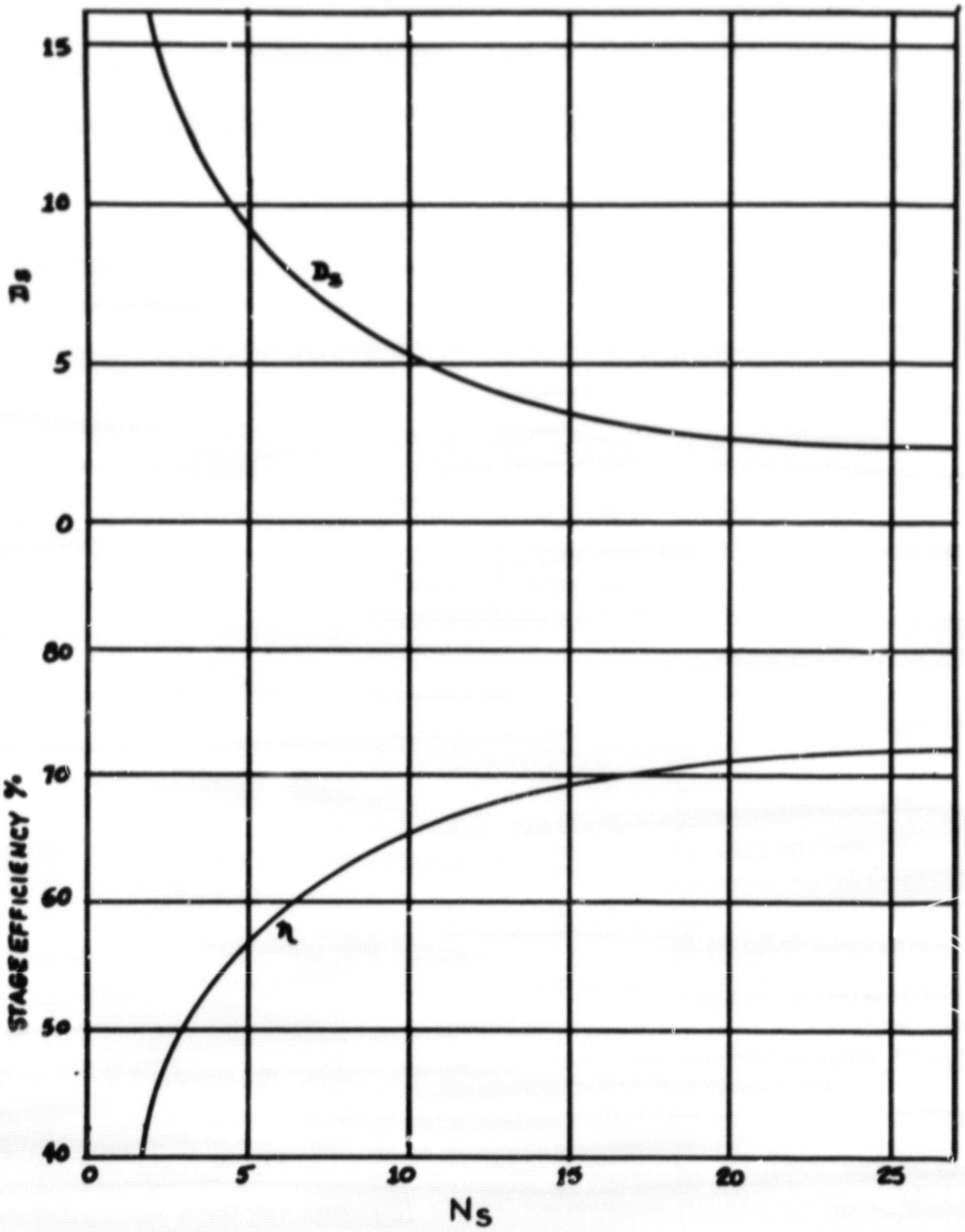

Fig. 13. Maximum Efficiency and the Corresponding D_s Versus N_s for Partial Admission Turbines. (From Balje, Ref. 21)

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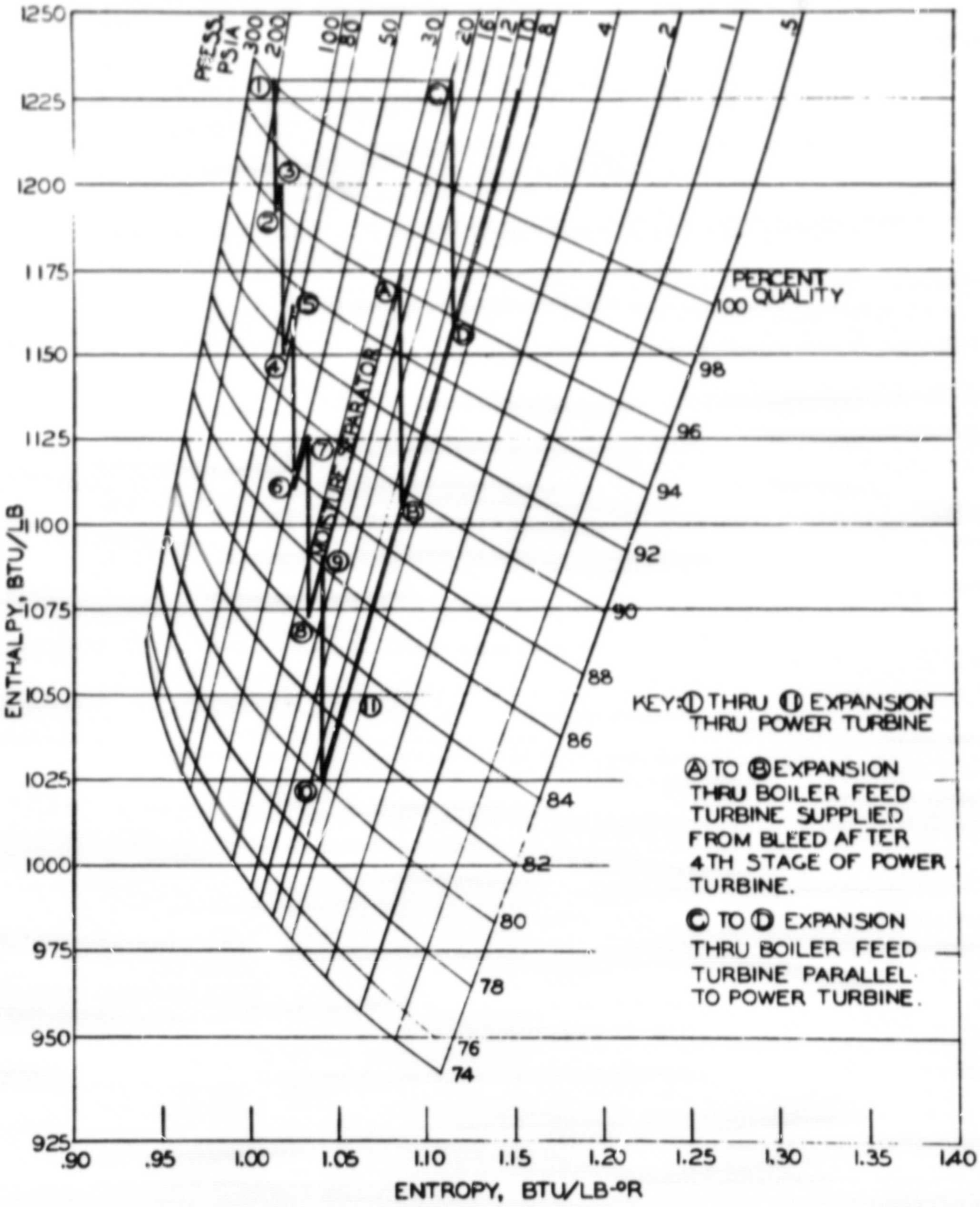

Fig. 14. Enthalpy-Entropy Diagram for Potassium Turbines. Data for potassium enthalpy and entropy taken from Ref. 22.

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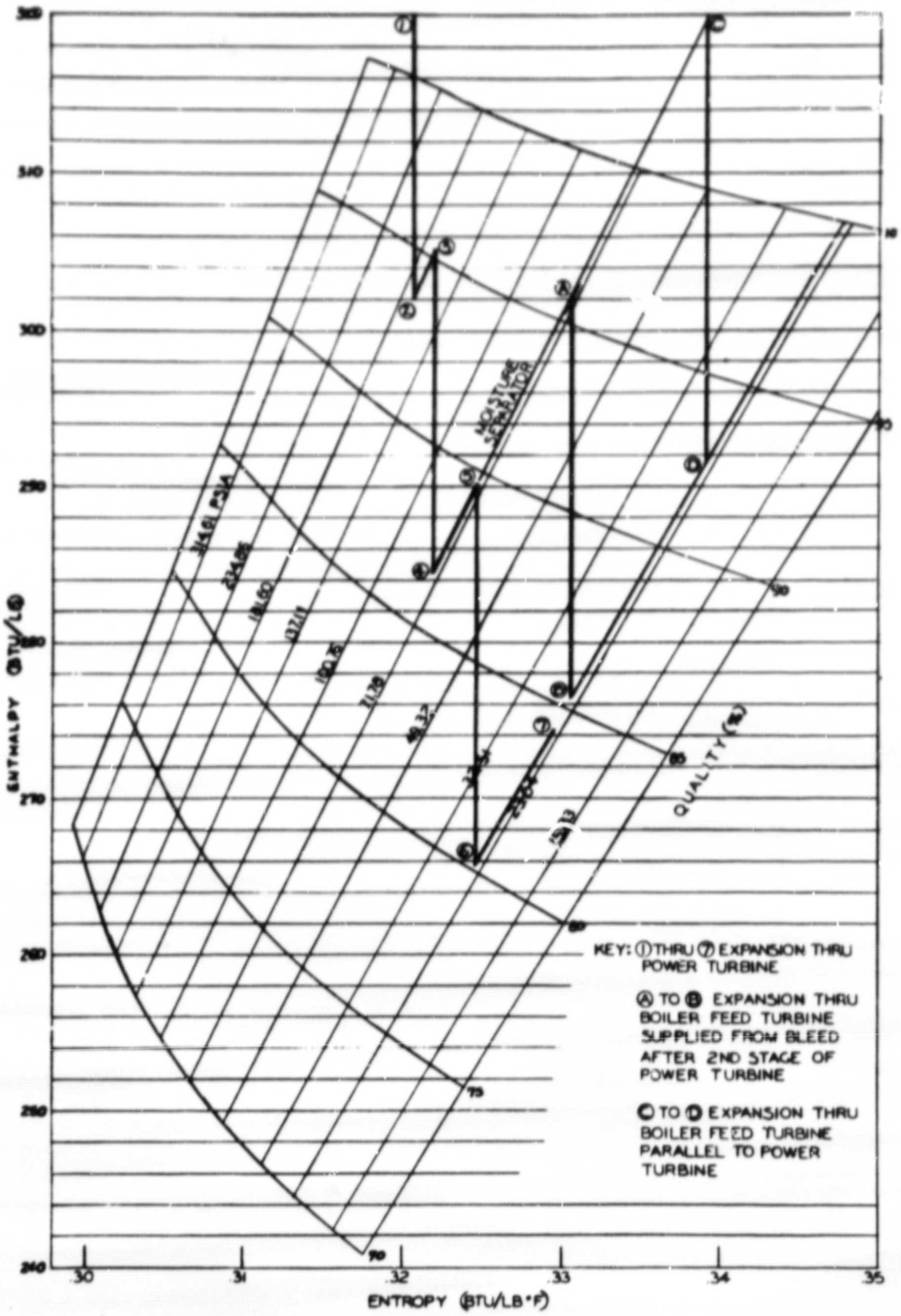

Fig. 15. Enthalpy-Entropy Diagram for Cesium Turbines. Data for cesium enthalpy and entropy taken from Ref. 23.

Table [illegible]. Summary of Pertinent Impeller Design Parameters for the Potassium and Cesium Boiler Feed Pumps at 1330°F

Parameters include hydraulic and cavitation conditions, jet pump characteristics, and impeller, jet pump, bearing, and windage power requirements.

Performance from Table 1			Impeller Parameters[a]					Cavitation Requirements												
														Jet Pump[b]						
Flow Q (gpm)	Head H (ft)	Head H (psi)	Speed N (rpm)	Specific Speed N_s	Specific Diameter D_s	Efficiency η (%)	Diameter D (in.)	NPSH/H	NPSH (ft)	NPSH (psi)	Suction Specific Speed S	Liquid Density (lb/ft^3)	Vapor Pressure (psia)	n	ϕ W_s/W_n	W_s (gpm)	W_n (gpm)	Assumed Bearing Flow (gpm)	Total Pump Flow Q_T (gpm)	Q_T/Q
												Potassium								
35.6	807	234.5	12,000	22	7.5	60	4.7	.024	19.2	5.6	7860	41.8	10.4	.024	6.5	23.7	3.6	6	33.3	.93
35.6	807	234.5	16,000	30	5.5	63	3.5	.033	27.0	7.8	8090	41.8	10.4	.033	5.1	23.7	4.6	6	34.1	.96
35.6	807	234.5	20,000	37	5.0	65	3.2	.044	36.0	10.5	8110	41.8	10.4	.044	4.2	23.7	5.6	6	35.3	.99
												Cesium								
60.5	534	334.6	12,000	40	4.5	66	4.1	.051	27.3	17.1	7840	90.2	23.6	.051	3.9	43.7	11.1	6	60.8	1.0
60.5	534	334.6	16,000	53	3.3	70	3.0	.075	40.0	25.0	7830	90.2	23.6	.075	3.0	43.7	14.6	6	64.3	1.06[d]
60.5	534	334.6	20,000	67	2.8	73	2.6	.101	53.9	33.7	7860	90.2	23.6	.101	2.3	43.7	19.0	6	68.7	1.13[d]

[a] From Balje´, Ref. 24.

[b] Nomenclature from Cunningham, Ref. 26.

[c] From Table 8.

[d] Jet pump flow requirement is larger than estimated in Table 1.

Table 7. Summary of Pertinent Impeller Design Parameters for the Potassium and Cesium Boiler Feed Pumps at 1330°F

Parameters include hydraulic and cavitation conditions, jet pump characteristics, and impeller, jet pump, bearing, and windage power requirements.

ler Parameters[a]			Cavitation Requirements														Turbine Power Requirement				
Specific Diameter D_s	Efficiency η (%)	Diameter D (in.)	NPSH/H	NPSH (ft)	NPSH (psi)	Suction Specific Speed S	Liquid Density (lb/ft^3)	Vapor Pressure (psia)	Jet Pump[b] n	ϕ W_s/W_n	W_s (gpm)	W_n (gpm)	Assumed Bearing Flow (gpm)	Total Pump Flow Q_T (gpm)	Q_T/Q	Hydraulic Power (kw)	Impeller Power Required (kw)	Bearing[c] and Windage Loss (kw)	Turbine Output (kw)	Turbine Output Corrected to Satisfy NPSH (kw)
							Potassium													
7.5	60	4.7	.024	19.2	5.6	7860	41.8	10.4	.024	6.5	23.7	3.6	6	33.3	.93	3.63	6.05	.11	6.16	6.16
5.5	63	3.5	.033	27.0	7.8	8090	41.8	10.4	.033	5.1	23.7	4.6	6	34.1	.96	3.63	5.85	.21	6.06	6.06
5.0	65	3.2	.044	36.0	10.5	8110	41.8	10.4	.044	4.2	23.7	5.6	6	35.3	.99	3.63	5.68	.33	6.01	6.01
							Cesium													
4.5	66	4.1	.051	27.3	17.1	7840	90.2	23.6	.051	3.9	43.7	11.1	6	60.8	1.0	8.78	13.30	.30	13.60	13.60
3.3	70	3.0	.075	40.0	25.0	7830	90.2	23.6	.075	3.0	43.7	14.6	6	64.3	1.06[d]	8.78	12.55	.48	13.03	13.80
2.8	73	2.6	.101	53.9	33.7	7860	90.2	23.6	.101	2.3	43.7	19.0	6	68.7	1.13[d]	8.78	12.04	.71	12.75	14.40

mated in Table 1.

Figures 16 and 17 show that the NPSH requirement increases almost linearly with speed. The relatively high vapor pressure of the condensate at 1330°F indicates that it might be practical to provide NPSH with subcooling. Figures 16 and 17 show the subcooling requirements as functions of pump speed and NPSH. Since cesium has roughly twice the vapor pressure and less than a third of the specific heat capacity of potassium, subcooling would appear somewhat more effective for cesium. However, the much higher mass flow rate and NPSH pressure requirement for cesium lead to considerably larger cooling loads. For example, at 12,000 rpm, 114 kw and 60 kw subcooling would be required with cesium and potassium, respectively, to provide the requisite NPSH without recourse to a jet pump.

Pump efficiencies, impeller diameters, and NPSH requirements were computed based on the pertinent information contained in Refs. 24–26.

Bearing Losses. A detailed bearing design study was not carried out for the turbine driven feed pumps. Instead, the friction losses in the journal and thrust bearings, as presented in Table 8, were calculated assuming hydrodynamic lubrication and bearing dimensions of 1 1/4-in.-diam by 1-in.-long with a diametral clearance of 2 mil.

Similarly sized journal bearings have been operated stably in potassium[11] at higher speeds than contemplated here; thus it appears reasonable to assume that a stable bearing can be designed for these conditions. Journal bearing friction losses were calculated using power loss equations in Ref. 27, and the results agreed with data in Refs. 28 and 29.

The thrust load is a function of the pressure differential between the pump inlet and the turbine cavity, the unbalanced hydraulic forces on the pump impeller, plus the axial force component from the turbine nozzles. Development tests with free turbine driven pumps at ORNL[10] provided experimental evidence that the value of the thrust force imposed by impeller hydraulic unbalance can be controlled. The NPSH values listed in Table 7 show that the greatest pressure differential between the turbine cavity and the pump inlet occurs for the 20,000 rpm cesium case, and would result in a thrust force of the order of 30 lb acting to move the shaft toward the turbine end of the unit. Assuming a thrust load of 30 lb and a 5 psi loading gave 1 1/4-in.-ID by 3-in.-OD dimension for the thrust

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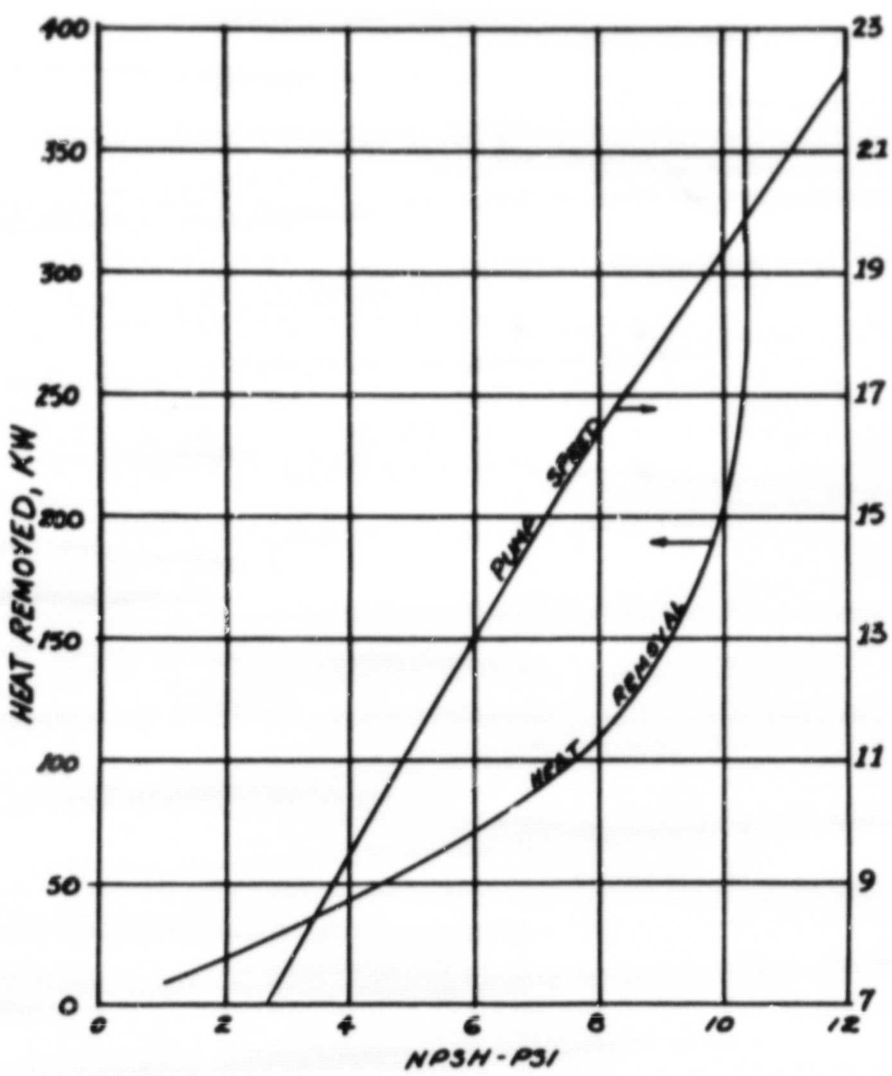

Fig. 16. NPSH Requirement Versus Speed for the Turbine-Driven Pump for the Potassium Boiler Feed Application and the Heat Removal Necessary to Provide the NPSH by Subcooling the Condensate Upstream of the Pump Inlet. Condenser outlet temperature 1330°F, vapor pressure 10.4 psia.

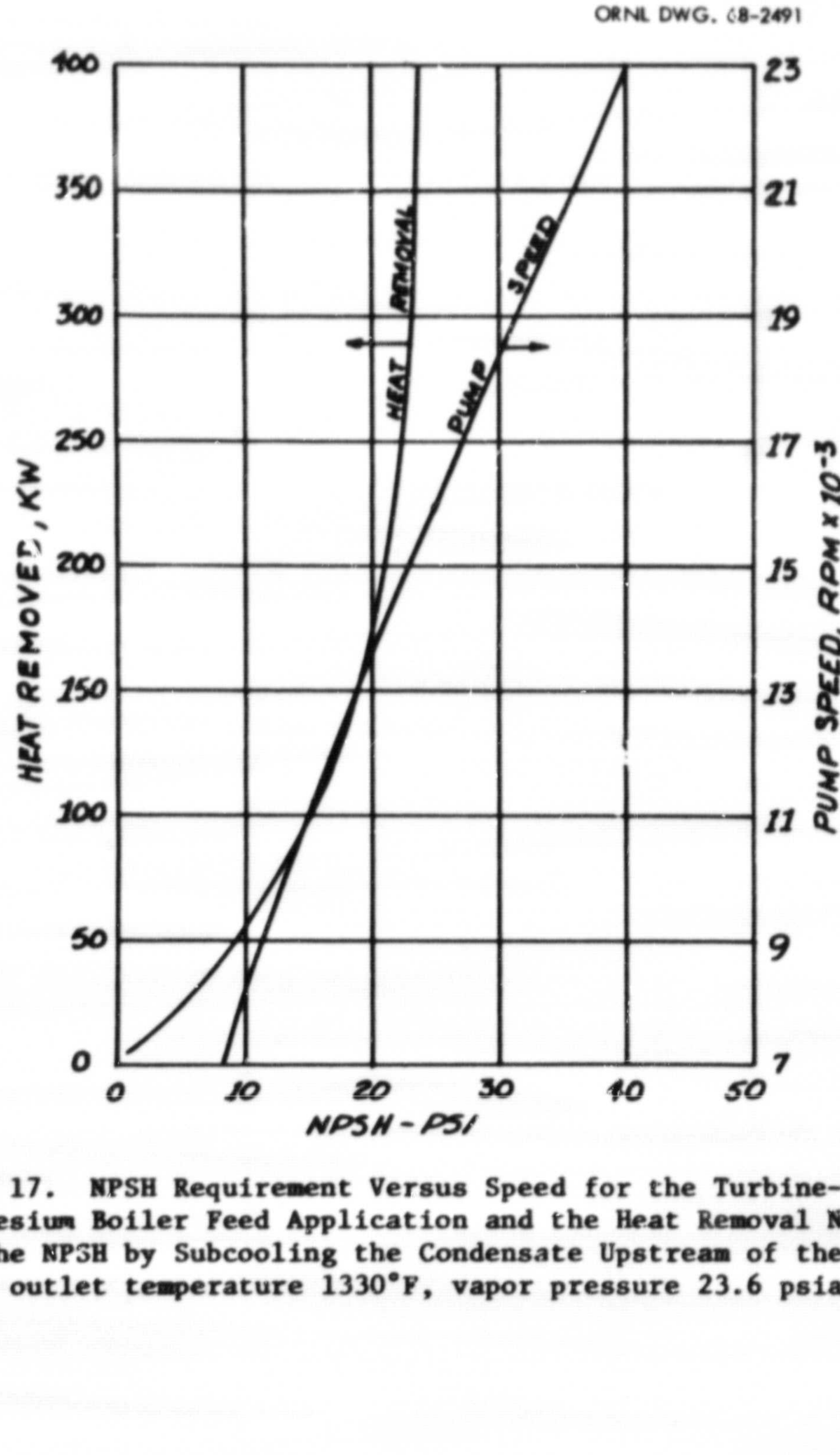

Fig. 17. NPSH Requirement Versus Speed for the Turbine-Driven Pump for the Cesium Boiler Feed Application and the Heat Removal Necessary to Provide the NPSH by Subcooling the Condensate Upstream of the Pump Inlet. Condenser outlet temperature 1330°F, vapor pressure 23.6 psia.

Table 8. Summary of Bearing and Windage Losses for the Potassium and Cesium Turbine-Driven Boiler Feed Pumps

		Journal Bearing								Thrust Bearing (4 pad pivoted bearing)						
Temperature (°F)	Density (lb/ft^3)	Viscosity (lb_f sec/$in.^2$ $\times 10^8$)	Journal Diameter (in.)	Journal Length (in.)	Radial Clearance (in.)	Shaft Speed (rpm)	Journal Reynolds No.	Critical Reynolds No.	Power Absorbed by one Bearing (kw)	Total Thrust Load lb_f	Unit Thrust Load ($lb/in.^2$)	d_2 (in.)	d_1 (in.)	d_{mean} (in.)	U (in./sec)	Minimum Film Thickness (in. $\times 10^6$)
									Potassium							
1330	41.8	1.99	1.125	1	.001	12,000	2480	1030	.016	30	5	3	1.125	2.14	1342	734
1330	41.8	1.99	1.125	1	.001	16,000	3300	1030	.036	30	5	3	1.125	2.14	1790	846
1330	41.8	1.99	1.125	1	.001	20,000	4120	1030	.053	30	5	3	1.125	2.14	2240	946
1200	42.8	2.12	1.125	1	.001	12,000	2370	1030	.017							
1200	42.8	2.12	1.125	1	.001	16,000	3160	1030	.032							
1200	42.8	2.12	1.125	1	.001	20,000	3960	1030	.054							
1040	44.1	2.39	1.125	1	.001	12,000	2170	1030	.018							
1040	44.1	2.39	1.125	1	.001	16,000	2900	1030	.034							
1040	44.1	2.39	1.125	1	.001	20,000	3620	1030	.059							
									Cesium							
1330	90.2	2.22	1.125	1	.001	12,000	4780	1030	.025	30	5	3	1.125	2.14	1342	775
1330	90.2	2.22	1.125	1	.001	16,000	6370	1030	.048	30	5	3	1.125	2.14	1790	895
1330	90.2	2.22	1.125	1	.001	20,000	7960	1030	.083	30	5	3	1.125	2.14	2240	999
1200	93.0	2.36	1.125	1	.001	12,000	4630	1030	.027							
1200	93.0	2.36	1.125	1	.001	16,000	6170	1030	.050							
1200	93.0	2.36	1.125	1	.001	20,000	7720	1030	.086							
1040	96.4	2.57	1.125	1	.001	12,000	4410	1030	.028							
1040	96.4	2.57	1.125	1	.001	16,000	5890	1030	.053							
1040	96.4	2.57	1.125	1	.001	20,000	7360	1030	.092							

Table 8. Summary of Bearing and Windage Losses for the Potassium and Cesium Turbine-Driven Boiler Feed Pumps

Journal Bearing							Thrust Bearing (4 pad pivoted bearing)										
ournal iameter (in.)	Journal Length (in.)	Radial Clearance (in.)	Shaft Speed (rpm)	Journal Reynolds No.	Critical Reynolds No.	Power Absorbed by one Bearing (kw)	Total Thrust Load lb_f	Unit Thrust Load (lb/in.^{2})	d_2 (in.)	d_1 (in.)	d_{mean} (in.)	U (in./sec)	Minimum Film Thickness (in. × 10^6)	Reynolds No.	Power Absorbed by one Bearing (kw)	Turbine Wheel Windage Loss (kw)	Specific Volume of Vapor (ft^3/lb)
						Potassium											
.125	1	.001	12,000	2480	1030	.016	30	5	3	1.125	2.14	1342	734	3270	.028	.009	43.6
.125	1	.001	16,000	3300	1030	.036	30	5	3	1.125	2.14	1790	846	5040	.041	.012	43.6
.125	1	.001	20,000	4120	1030	.053	30	5	3	1.125	2.14	2240	946	7040	.058	.017	43.6
.125	1	.001	12,000	2370	1030	.017											
.125	1	.001	16,000	3160	1030	.032											
.125	1	.001	20,000	3960	1030	.054											
.125	1	.001	12,000	2170	1030	.018											
.125	1	.001	16,000	2900	1030	.034											
.125	1	.001	20,000	3620	1030	.059											
						Cesium											
.125	1	.001	12,000	4780	1030	.025	30	5	3	1.125	2.14	1342	775	6350	.031	.072	5.57
.125	1	.001	16,000	6370	1030	.048	30	5	3	1.125	2.14	1790	895	9760	.048	.096	5.57
.125	1	.001	20,000	7960	1030	.083	30	5	3	1.125	2.14	2240	999	13600	.069	.121	5.57
.125	1	.001	12,000	4630	1030	.027											
.125	1	.001	16,000	6170	1030	.050											
.125	1	.001	20,000	7720	1030	.086											
.125	1	.001	12,000	4410	1030	.028											
.125	1	.001	16,000	5890	1030	.053											
125	1	.001	20,000	7360	1030	.092											

bearing. Since the NPSH requirement and developed pressures are lower for potassium, it is likely that the assumed thrust loading is very conservative for potassium.

The power absorbed by the thrust bearing varies approximately with the thrust load and the area of the thrust face and was calculated on the basis of experimental data on NaK lubricated thrust bearings.[30] These data correlated well with the results of thrust bearing tests in water.[31]

The power required by the journal and thrust bearings is given in Table 8. In determining the power output required from the turbine wheel the sum of the losses for two journals and one thrust bearing was increased to double the values shown in Table 8, to compensate for the high values of the bearing Reynolds numbers, all of which are above the critical value.

Windage Loss and Disk Friction. The disk friction and windage loss for the partial admission, single-stage, turbine wheel were calculated according to Sternlicht[27] and are listed in Table 8. The loss is inversely proportional to the specific volume of the exhaust vapor so that the denser cesium vapor has larger friction losses.

COMPARISON OF BOILER FEED PUMPS

Pump Weights and Weight Penalties

The three boiler feed pumps for potassium and cesium are compared in Table 9 on the bases of (1) the weight of the basic pump, (2) a weight penalty for the electrical power consumed by the pump, and (3) the weight penalties for the auxiliaries required to provide pump control, cooling, power conditioning, and start-up. For the free turbine driven pump, the equivalent electrical power that could have been generated with the vapor that must be diverted to the drive turbine was used to compute the electrical power penalty. A weight penalty was also assessed for the batteries required to start up the helical induction and the canned rotor pumps. Table 10 presents a comparison of battery systems. No attempt was made to compare the pumps on a cost basis.

Table 9. Weight Comparison of the Helical Induction, Canned-Rotor, and Free Turbine-Driven Pumps Applied to the Potassium and Cesium Boiler Feed Requirements.

Comparison based on (1) pump weight, (2) weight and power penalties, and (3) weight, penalties, and batteries.

	Potassium (1330°F)				Cesium (1330°F)			
Pump Type	Electromagnetic Helical Induction	Centrifugal	Centrifugal	Centrifugal	Electromagnetic Helical Induction	Centrifugal	Centrifugal	Centrifugal
Pump Drive	---	Canned Rotor	Free[a,b] Turbine	Free[a,b] Turbine	---	Canned Rotor	Free[a,b] Turbine	Free[a,b] Turbine
Pump flow, gpm	30.5	35.6	35.6	35.6	52.9	60.5	60.5	68.7[f]
Pump head, ft	807	807	807	807	534	534	534	534
Speed, rpm	---	12,000	12,000	20,000	---	12,000	12,000	20,000
Turbine wheel diameter, in.	---	---	12.0	7.4	---	---	8.6	5.5
Impeller diameter, in.	---	4.7	4.7	3.2	---	4.1	4.1	2.6
NPSH required, ft	8	19	19	36	6	27	27	54
Impeller efficiency, %	---	---	60	65	---	---	66	73
Overall efficiency, %	19.5	30	33.5	37	15	32	36.4	40.6
Power input, kw	16	12.1	6.91[c]	6.26[c]	51.5	27.4	13.7[c]	13.9[c]
Power input as % of system generator capacity	4.86	3.67	2.1	1.9	15.8	8.4	4.2	4.26
Basic pump weight, lb	397	160	147	72	1430	274	85	35
Consumed power weight penalty at 10 lb/kw, lb[d]	160	121	69	63	515	274	137	139
Power conditioning weight penalty, lb	16	12	0	0	52	27	0	0
Cooling equipment weight penalty, lb	24	18	0	0	77	41	0	0
Reactive power weight penalty, lb	23	6	0	0	89	13	0	0
Total - pump weight plus weight penalties, lb	620	317	216	135	2163	629	222	174
Batteries for start-up, lb[e]	40	30	0	0	129	68	0	0
Total weight pump, penalties, and batteries, lb	660	347	216	135	2292	697	222	174
Total weight as percent of heaviest pump	100	54.6	34	21.2	100	34.6	11	8.6
Total weight pump, penalties, and batteries based on consumed power penalty of 25 lb/kw, lb	969	552	310	228	3110	1229	427	382
Total weight as percent of heaviest pump	100	57	32	23.5	100	39.4	13.7	12.3

[a] Data shown are for boiler feed drive turbine supplied by inter stage bleed from power turbine (Fig. 12).

[b] Increase total pump weight plus weight penalty values shown by 50 to 80% for boiler feed drive turbine arranged parallel to power turbine (Fig. 11).

[c] Vapor flow rate of boiler feed turbine converted to equivalent electrical output of turbine-generator. (See footnotes Table 4.)

[d] See Ref. 5 for basis for system weight penalties.

[e] Assuming Ag-Zn batteries and 30 minutes to reach half power at which time the turbine-generator supplies all power. (See Table 10.)

[f] Pump flow was increased to supply jet pump flow to meet NPSH requirement.

Table 10. Start-up Battery Parameters: Energy Density, Life, and Cost[32]

	Lead-Acid	Nickel-Iron	Nickel-Cadmium	Silver-Zinc	Silver-Cadmium	Zinc-Air
Energy density						
Theoretical, whr/lb	115	215	99	220	134	400
Actual, whr/lb	12	12	16	50	30	50
whr/in.^{3}	0.9	0.8	1.6	3.0	2.0	2.5
ft^3/kwhr	0.65	0.72	0.36	0.20	0.29	0.23
lb/kwhr	83	83	63	20	33	20
Cycle life, number of cycles	500–2000		1800	100–150	1500	
Initial cost, $/kwhr	65			240–460		
Salvage value, % of initial value	7			15–33		
Total life, kwhr/lb	15		29	6	45	
Total battery cost,[a] mills/kwhr	50			1900[b]		

[a]Initial cost less salvage value divided by total life in kwhr.

[b]Silver-zinc battery cost based on $300/kwhr capacity and 25% salvage value.

Uncertainty exists about the value of the weight penalty that should be assigned to the electrical power consumed by the pumps. An earlier study[8] used a weight penalty of 10 lb/kw. A typical Rankine cycle system with an electrical output of 330 kwe may weigh in the order of 8500 lb, that is, 26 lb/kw without shielding. The 10 lb/kw value was used, but the ranking of the pumps as presented in Table 9 is the same for either value of the pump power weight penalty. The helical induction pump has both the largest basic pump weight and the largest sum of weight penalties, while the free turbine driven pump has the lowest values for these two parameters. The canned rotor pump ranks about midway between the other two.

The canned rotor pump appears to be inherently the most complex, because it requires both the liquid metal lubricated bearings needed with the free turbine driven pump and the power supply, control, and conditioning, heat removal and battery start-up systems needed with the helical induction pump.

Special Problems

Variable Frequency Control and Start-up of Electromagnetic and Canned Rotor Pumps

A variable frequency system can provide smooth, stepless control for either the helical induction or the canned rotor pumps and should be suitable for all anticipated operational conditions. Stepless control can be provided by a cycloconverter, a frequency step-down device, that can be supplied from the high frequency (of the order of 1000 to 3000 cps) bus of the system turbine-generator. The cycloconverter is comprised of a number of solid state switches (probably silicon-controlled rectifier units) connected between the power system and the pump as shown in Fig. 19. Logic circuitry based on pertinent parameters will be used to control the output frequency and voltage which in turn control pump performance. Additional information on the system is given in Refs. 9 and 14.

A step-down transformer between the power supply and the cycloconverter will be needed if the generator voltage is greater than 400 volts.

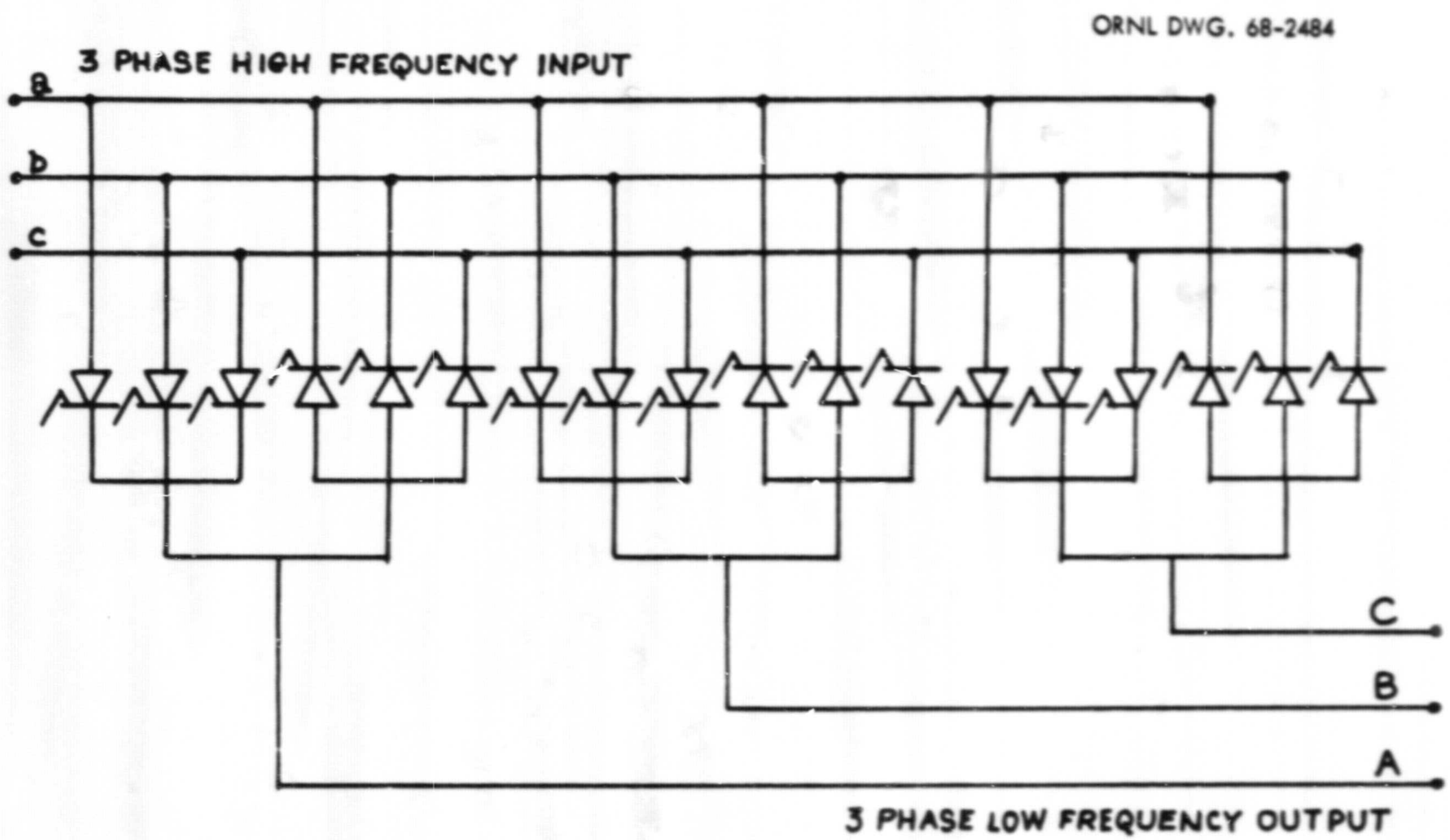

Fig. 18. Electrical Diagram for the Cycloconverter Frequency Divider. Triggering and gating circuits are not shown.

It was assumed that the size of this transformer will be sufficient to take care of all low voltage loads. An additional weight penalty of 2 lb/kw for a generator frequency of 1000 cps* must be assessed against both the helical induction and the canned-rotor pumps if this transformer is required.

Power for the startup of the helical induction pump and the canned rotor pump was assumed to be supplied by a battery system. The dc output of the battery would be changed with a suitable solid state inverter to three-phase ac at about the same frequency as the normal output of the system turbine generator. The output of the inverter would be connected to the input (high frequency) of the cycloconverter frequency divider used for the steady-state control of these pumps.

<u>Control of Free Turbine-Driven Boiler Feed Pumps</u>

<u>Boiler Feed Turbine in Parallel with the Power Turbine</u>. A flow schematic of the feed pump installed in the parallel arrangement is shown in Fig. 11. Some of the methods of matching pump flow-rate to the flow requirements of the Rankine cycle system include (1) throttling the vapor flow at the turbine inlet, (2) throttling the pump flow, and (3) using cavitation control at the feed pump inlet to provide the proper flow-rate and to maintain the proper distribution of liquid inventory in the system without throttling either the vapor or the liquid flow streams.

The various methods of flow control have differing influences on the coupling between the pump and the power turbines. The vapor flow-rate to the pump drive turbine is less than 10% of that to the power turbine; therefore throttling the vapor flow at the inlet to the pump drive turbine should have small influence on the performance of the power turbine. Throttling the pump flow rate should have only a small effect on the speed of the feed pump and little or no effect on the performance of the power turbine. The influence of cavitation control on the coupling is discussed below.

The cavitation control method depends primarily upon a mismatch between the non-cavitating capacity of the feed pump and the flow requirements of the Rankine cycle system, that is, the pump operates at slightly higher speed than necessary. The non-cavitating capacity is higher than the requirement, so the pump operates in a regime of cavitation to supply the precise system requirement. The control method will function properly over whatever power range the non-cavitating capacity of the feed pump exceeds the system requirements.

In tests with simulated Rankine cycle systems at ORNL in which the power turbine was simulated by a fixed vapor flow resistance, the cavitation control was operable over the power range from 25 to 100% of the design value. Throughout the range there was no need to throttle either the vapor or the liquid flow streams. The flow resistances must be set initially to the proper values. These tests were conducted in both water and potassium systems of 360 kw boiler capacity.

The parallel arrangement of pump and simulated power turbine has been used repeatedly to perform the boot-strap start-up of a 360 kw water Rankine cycle system without the use of any auxiliary system.

Boiler Feed Turbine Pump Supplied with Interstage Bleed Vapor from the Power Turbine. The flow schematic of a feed pump turbine supplied with vapor bled from the power turbine is shown in Fig. 12. This pump arrangement was adopted for the comparison because it provided the most efficient use of working fluid vapor of the three arrangements considered. We have had no test experience with this pump arrangement in the ORNL simulated Rankine cycle systems, but believe that it should be amenable to the same control methods used with the parallel arrangement described in the preceding section. Those methods include (1) throttling the vapor flow at the turbine inlet, (2) throttling the pump flow, and (3) using cavitation control at the feed pump inlet.

There is somewhat more coupling between the pump drive turbine and the power turbine in the bleed vapor arrangement than with the parallel arrangement. At full-power the vapor flow-rate to the feed pump turbine is less than 10% of that through the power turbine; therefore, one would expect changes of as much as 10% in the vapor flow-rate to the feed pump turbine to have little effect on the performance of the power turbine. Throttling the pump flow should have little or no effect on power turbine performance. Precise control of flow-rate matching and the distribution of system liquid inventory can be obtained with the cavitation method of control, but the power range over which it is effective without the use of throttling may be smaller than for the parallel arrangement.

To provide for operation at low system power and for boot-strap start-up of the system, it is necessary to utilize the two vapor valves and the bypass line direct from the boiler to a separate nozzle block installed in the feed pump turbine shown in Fig. 12.

Bearing Materials

The operational reliability of the liquid metal lubricated bearings which support and position the rotating elements in the power turbine and its associated electrical generator may well determine the useful life of the Rankine cycle system in a space power application.

Hydrodynamic bearings, lubricated with liquid potassium at temperatures to 1040°F, have been used to support and position the rotating element of free turbine driven boiler feed pumps operated in simulated Rankine cycle systems at ORNL and more than 8,000 hr of turbine running time has accumulated in these systems. A typical turbine pump unit used in these systems and shown in Fig. 19 is described in Ref. 10. The bearings were initially made of tungsten carbide with 12% cobalt binder, which composition was subsequently changed to tungsten carbide with 6% cobalt binder. There is some question about the use of either of these bearing materials at the condensate temperature of 1330°F considered in this report or for any temperature above about 1200°F.

A materials program for potassium lubricated journal bearings has been conducted by R. G. Frank et al at General Electric-Evendale under contract to NASA.[33–3] The results of the program indicate that titanium carbide with 10% columbium binder is a good candidate for both journals and bearings. The material reportedly has good compatibility with potassium up to 1600°F in columbium, tantalum, and tungsten base alloys and also has good general bearing properties. The primary problem with the material appears to be the exercise of appropriate quality control during the fabrication processes.

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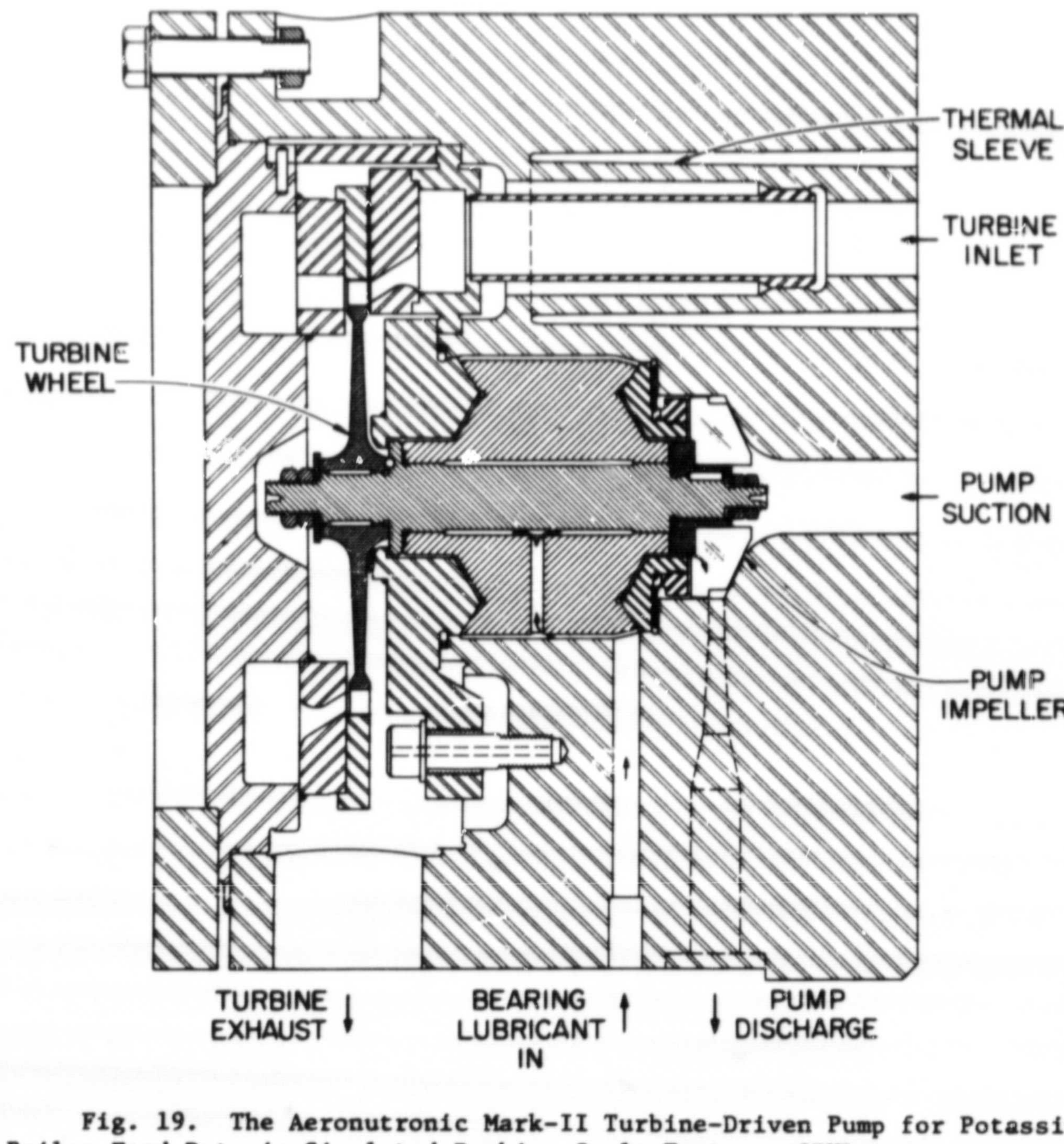

Fig. 19. The Aeronutronic Mark-II Turbine-Driven Pump for Potassium Boiler Feed Duty in Simulated Rankine Cycle Tests at ORNL.

PRELIMINARY DESIGN OF ELECTROMAGNETIC PUMPS FOR THE REACTOR COOLANT AND HEAT REJECTION SYSTEMS

The lithium and NaK working fluids, respectively, in the reactor coolant and heat rejection systems of the reference space power systems of Ref. 1, are used in the liquid phase only. A brief study of the application of the free turbine driven pump to these systems indicated that it would lead to increased problems and unwanted complexity. Supplying potassium vapor to the lithium and NaK turbines from the Rankine cycle system would give rise to unwanted communication among the three systems or to severe shaft sealing problems. Boiling lithium or NaK to provide vapor for turbines would require individual boiling and condensing capabilities and lead to significant increases in system complexity and probably to increases in weight compared to the electromagnetic pump.

The flat linear induction pump (FLIP) and the annular induction pump (AIP) were selected to pump the lithium and NaK, respectively, for reasons noted below. The preliminary designs of the pumps are shown in cross-section in Figs. 20 and 21, and their performance characteristics are shown in Table 11. The relatively high NPSH requirements are caused mainly by the relatively high duct velocities of 30 and 40 ft/sec that were used.

Lithium Pump for the Reactor Coolant System

The high flow, low head (374 gpm at 20 psi) requirements of this pump are suited to either the annular induction pump (AIP) and the flat linear induction pump (FLIP). The FLIP in which all the magnetic circuit is readily accessible for cooling is particularly well suited to this application where the liquid temperature is 2200°F. The use of the annular induction pump would require separate liquid metal cooling of the inner core to maintain its temperature below the Curie temperature.

The design of the flat linear induction pump for this service is similar to that of the pump discussed on pages 133–135 and 234 of Ref. 5. The pump duct is made of D43 alloy and is approximately 0.75-in. by 6-in.

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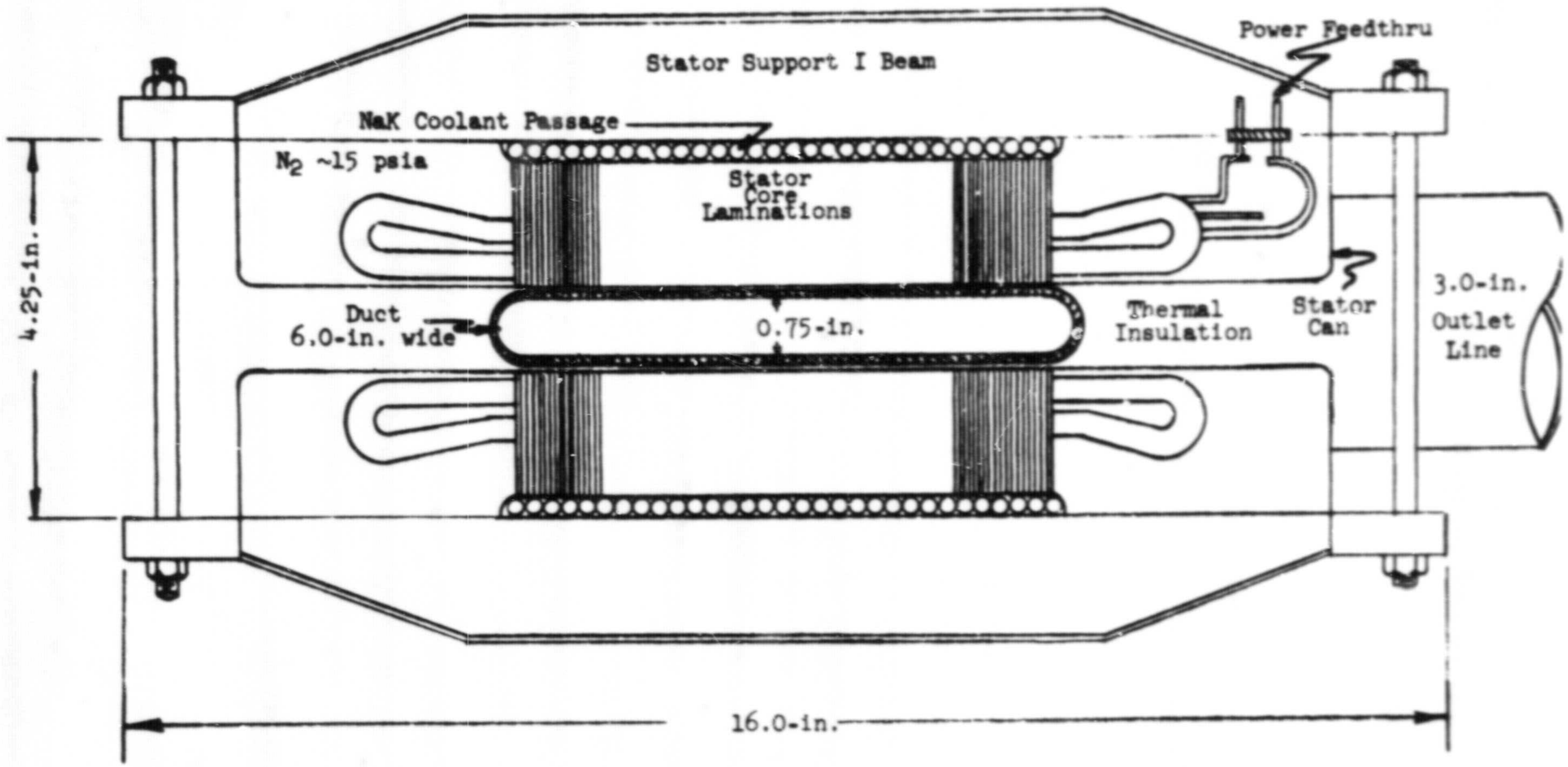

Fig. 20. Dross Section of the Preliminary Design of the Flat Linear Induction Pump (FLIP) for Lithium in the Reactor Coolant System. Basic pump weight 378 lb.

Notes:
1. Stator core and windings are 36.0-in. in length.
2. 3.0-in. inlet is at front end of pump, direction of flow is into plane of paper.
3. Stator support I-beams are at 9.0-in. intervals.

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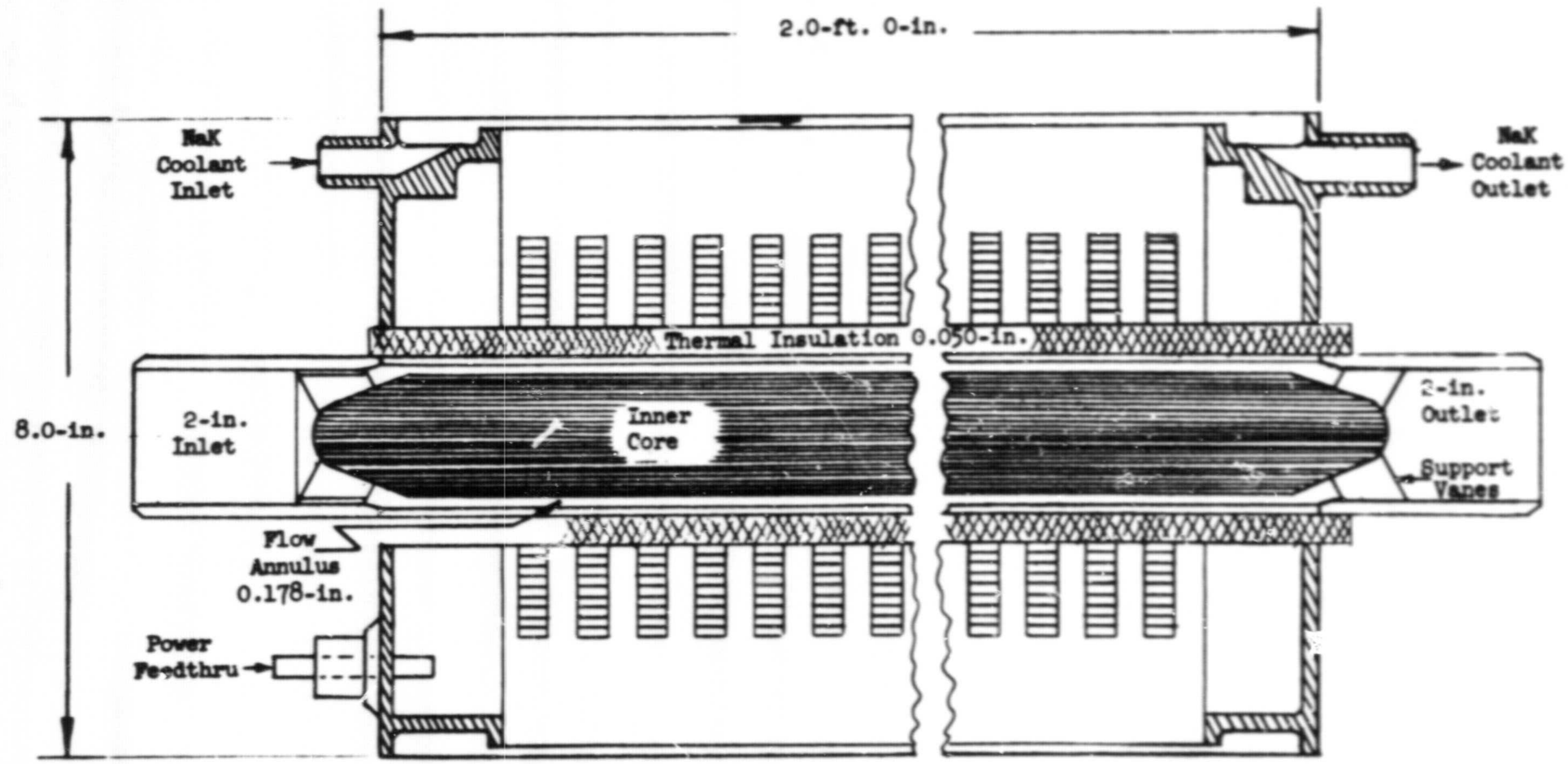

Fig. 21. Cross Section of the Preliminary Design of the Annular Induction Pump (AIP) for NaK in the Heat Rejection System. Basic pump weight 330 lb.

Table 11. Characteristics of Electromagnetic Pumps for the Reactor Coolant (Lithium) and Heat Rejection (NaK) Systems in a 330 Kwe Space Power Plant.

Flat linear induction pump (FLIP) for lithium and annular induction pump (AIP) for NaK.

Item	Lithium Coolant (FLIP)	NaK Coolant (AIP)	Remarks
1. Flow, lb/sec	22	13.6	
2. Flow, gpm	374	136	
3. Head, psi	20	20	
4. Head, ft	109	64	
5. Liquid metal temperature, °F	2200	1200	
6. Density, lb/ft^3	26.45	45	
7. Resistivity, microhm-in.	21.85	32.7	
8. Pump potential, volts	150	200	
9. NPSH, ft	28	50	
10. PCP $\times 10^{-3}$	162	122.4	
11. Pump frequency, cps	100	200	
12. Pump output, kw	3.25	1.18	
13. Pump power input, kw	27.0	11.8	
14. Pump efficiency, %	12	10	
15. Pump input, kva	53.0	33.7	
16. Pump P.F., %	51	35	
17. Pump reactive input, kvar	45.6	31.6	
18. Base pump weight, lb	378	330	
19. Consumed power, weight penalty, lb	270	118	(10 lb/kw- Item 13)
20. Power conditioning, weight penalty, lb	27	12	(1 lb/kw- Item 13
21. Cooling equipment, weight penalty, lb	41	18	(1.5 lb/kw- Item 13)
22. Reactive power, weight penalty, lb	34	24	(0.75/kvar - Item 17)
23. Pump weight + weight penalties, lb	750	502	

by 36-in. long. The fluid velocity is 30 ft/sec at design flow. The inlet and outlet lines are 3-in.-diam tubes at right angles to the pump duct. These lines connect to the flow transition sections which are offset cones. A cross section of the pump is shown in Fig. 20.

The pump duct is not mechanically connected to the stator but is supported by the thermal insulation which protects the stator from high duct temperatures. Duct guides are required to limit the lateral displacement. Thermal insulation consists of layers of tantalum cloth and strip with an overall depth of 0.15-in. The outer layers of insulation can be austenitic stainless steel.

The stator consists of two identical stator assemblies held together by modified I-beams located at 9-in. intervals along the length of the duct. A NaK coolant jacket covers the outer surface of the hermetically sealed stator can. This stator can is a metallic envelope containing approximately one atmosphere of nitrogen. Nickel-clad silver conductors with inorganic insulation and Hiperco 27 laminations with plasma-sprayed alumina coatings are used for the stator windings and core, respectively. The stator is 6 pole, 3 phase, 100 cps.

NaK Pump for the Heat Rejection System

The operational requirements for this pump, Fig. 21, are very similar to those for a similar pump whose design is covered in pages 123–128 of Ref. 5, in which the annular induction pump was a narrow choice over the helical induction pump. The margin is larger for the NaK circuit pumps since the higher flow, approximately twice that of the pump in Ref. 5, is more suited to the annular induction pump. The weight and dimensions of the pump were determined in part from curve V in Fig. 4, and in part from an analytical extrapolation of the Ref. 5 pump to the requirement of the heat rejection circuit. The efficiency and power factor were changed to agree with present day state of the art.

The pump duct and all other NaK wetted parts of the pump may be fabricated from an austenitic stainless steel. The inner core is an assembly of axially oriented square wires of cobalt iron alloy. This core is encased in the stainless steel and is supported by four radial support vanes at each end of the pump cell. The annular flow passage between the inner core and the pump duct has a radial dimension of 0.173-in. Fluid velocity in the annulus is approximately 40 ft per second at design conditions. The pump duct is supported within the stator by the 50-mil-thick thermal insulation consisting of alternate layers of austenitic stainless steel cloth and strip that is laminated to reduce eddy current loss.

The stator is 2 pole, 3 phase, 200 cps and uses NaK (700°F maximum) in the cooling jacket. The stator cavity is hermetically sealed and is filled with one atmosphere of nitrogen. Nickel-clad silver conductors with inorganic insulation and Hiperco 27 laminations with plasma-sprayed alumina coatings are used for the stator windings and core, respectively. Electrical power to the stator is supplied through three hermetically sealed ceramic-to-metal electrodes located at one end of the stator cavity.

CONCLUSIONS

1. Of the several variations of the electromagnetic pump considered for the boiler feed applications, the polyphase helical induction pump (HIP) was chosen as having the most desirable features.

2. When compared to potassium, the use of cesium as the Rankine cycle working fluid requires much heavier boiler feed pumps as shown by weight (lb):

Pump Kind	Cesium		Potassium	
	Basic Pump	Total Pump Plus Penalty	Basic Pump	Total Pump Plus Penalty
Helical induction polyphase	1430	2292	397	660
Canned rotor centrifugal	274	697	160	347
Free turbine driven centrifugal	35	174	72	135

3. When compared for either the cesium or potassium boiler feed applications, the helical induction pump is the heaviest, the free turbine driven pump is the lightest, and the canned rotor pump is intermediate in weight.

4. The helical induction pump which has no moving parts can be designed to be relatively free of thermal stresses. However, it requires an electrical power supply, electrical switchgear and control equipment, auxiliary cooling equipment, and starting batteries.

5. Although the free turbine driven pump requires no electrical power supply and little or no auxiliary equipment, it depends upon moving parts and liquid metal lubricated bearings. Development tests will be required to prove the bearing materials selection and the reliable and stable operation of the bearings.

6. The canned rotor pump requires liquid metal lubricated bearings, similar to those used in the free turbine driven pump, and the electrical power supply, electrical equipment, auxiliary cooling equipment and starting batteries, similar to those used in the helical induction pump. Thus, although the canned rotor pump is intermediate in weight, it is the most complex of the three boiler feed pumps considered.

7. Of the various arrangements for incorporating free turbine driven feed pumps in a Rankine cycle system, the parallel arrangement of the power and feed pump turbines shown in Fig. 11 provides the largest decoupling of the effects of a change in the operating conditions of the pump drive turbine on the performance of the power turbine. The stage bleed arrangement shown in Fig. 12 provides somewhat closer coupling than the parallel arrangement but yields the minimum system weight for any boiler feed pump (30 to 40% below the weight for the parallel arrangement).

8. The higher density of the cesium and the higher specific speed for the cesium pump lead to a higher net positive suction head pressure requirement (NPSH) for the cesium than for the potassium centrifugal pump.

9. Subcooling the condensate imposes a larger weight penalty than the use of jet pumps to provide the net positive suction head requirement. However, the jet pumps utilize high nozzle velocities that may require special designs to avoid erosion.

10. Cavitation at the inlet to the centrifugal pump provides an excellent control method for matching the capacity of the feed pump to the Rankine cycle requirement and for controlling the distribution of liquid metal inventory in the system. This control method has been used with free turbine driven boiler feed pumps during more than 5,000 hr of test operation in simulated Rankine cycle systems at ORNL (2,700 hr on a single pump unit) with no visual evidence of cavitation damage to an impeller.

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